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NOTES

ON

STEAM BOILERS

PREPARED FOR THE USE OF STUDENTS
AT THE MASSACHUSETTS INSTITUTE OF TECHNOLOGY

BY

C. H. PEABODY AND E. F. MILLER

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PREFACE.

THESE notes on Steam Boilers have been prepared for the use of students at the Massachusetts Institute of Technology. They represent part of the course of instruction in steam engineering, which makes a study of valve gears, the steam-engine indicator, thermodynamics, steam engines, and steam boilers, and which is accompanied by a course in applied mechanics and strength of materials. Chapter XIV. was prepared by Professor Schwamb.

A part of the illustrations were made for this work, but the larger part were loaned to the Institute by the Hartford Steam Boiler Inspection and Insurance Company, by the Crosby Gauge and Valve Company, by Hilles and Jones, by Bement, Miles, and Company, by the Niles Tool Works, and by the makers of boilers, boiler fitting, and apparatus, whose names are given in connection with the descriptive matter.

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STEAM BOILERS.

* * *

CHAPTER I.

TYPES OF BOILERS.

STEAM boilers may be classified according to their form and construction, or according to their use. Thus we have horizontal and vertical boilers, internally and externally fired boilers, shell boilers and sectional boilers, fire-tube and water-tube boilers; the several features mentioned may be combined in various ways, so as to give rise to a large number of kinds and forms of boilers. Again, we have stationary, locomotive, and marine boilers, together with a variety of portable and semi-portable boilers. Locomotive boilers are always shell boilers, internally fired, and with fire tubes; and the restrictions of the service have developed a form that has changed little from the beginning, except in the direction of increased size and power. Marine boilers present a much larger variety of form and construction, yet present practice in marine engineering gives preference to the Scotch, or drum, boiler. The variety of forms and methods of construction of stationary boilers is very wide; each country and section of a country is likely to have its own favorite type. Thus, in New England, where the water is good, cylindrical tubular boilers are largely used; in some of the western states, where water contains mineral impurities, flue boilers are preferred; and in England, the Lancashire and Galloway boilers are favored; and again, various forms of sectional and water-tube boilers have come into use, especially in later years, with the increase of steam pressures.

Cylindrical Tubular Boiler. — This type of boiler is shown by Fig. 1 and by Plate I., and also with setting in brick work, by Plates II. and III. It consists essentially of a cylindrical shell closed at the ends by two flat *tube plates,* and of numerous *fire tubes,* commonly having a diameter of four inches. About two-thirds of the volume of the boiler is filled with water, the other third being reserved for steam. The water line is six or eight inches above the top row of tubes. The tube plates below the water line are sufficiently stayed by the tubes; above the water line the flat plates are stayed by *through rods* or *stays* as in

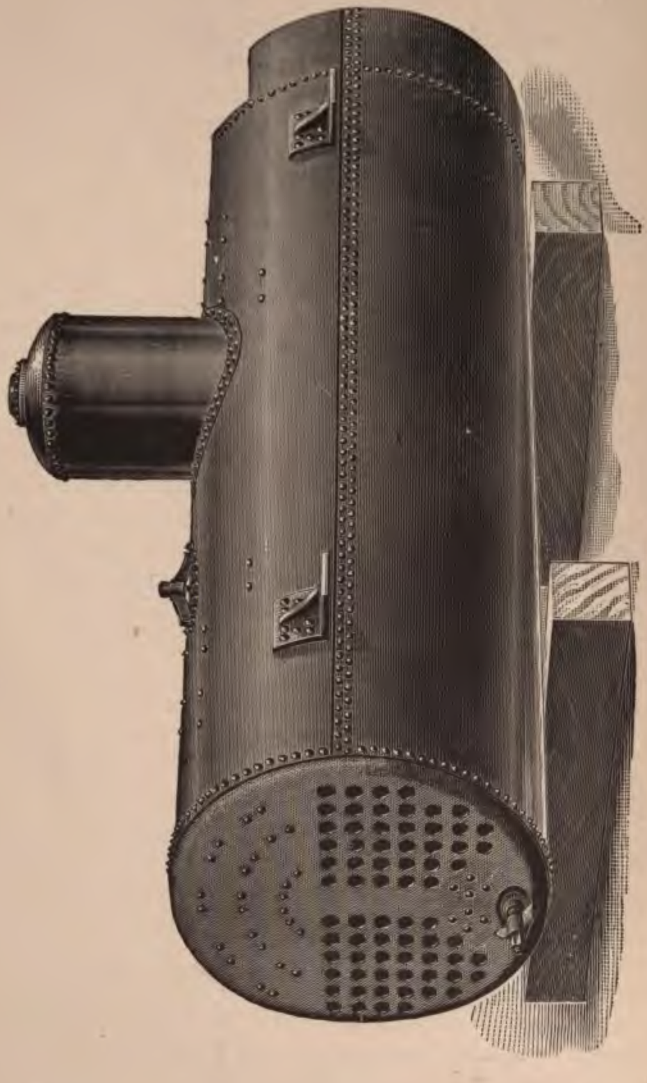

FIG. 1.

Plate I., by diagonal stays like those shown in Fig. 12, or otherwise. The furnace is under the *front end* of the boiler, as shown in Plates II. and III., and the products of combustion pass back, over a wall or *bridge* limiting the furnace, to the *back end* (shown at the left-hand end of Fig. 1), then forward through the tubes and up the *uptake* or flue leading to the chimney. The shell commonly protrudes beyond the front tube plate, and is cut away at the top to facilitate the arrangement of the uptake. The boiler is supported by cast-iron brackets, riveted to the shell; the front brackets may rest on, or be fixed to, the supporting side walls, but the rear brackets should be given some freedom, so that the boiler may not be unduly strained by expansion. For the same reason space is left between the back tube plate and arch. Plates II. and III. show the rear brackets provided with rollers and iron bearing plates. Figure 1 and Plates II. and III. show a vertical *steam drum* on top of the boiler, from which steam is drawn. Plate I. shows only two steam nozzles near the ends of the boiler. The boiler is provided with a *manhole* on top and *hand holes* near the bottom at the ends, to give access to the interior of the boiler, and facilitate washing out. Plate I. shows the *feed pipe,* entering just in front of the rear steam nozzle; it drops down, and then has two horizontal branches perforated on top to distribute the water. The same plate shows an opening in the bottom near the rear tube plate for the *blow-out pipe,* used in emptying the boiler. About half the shell, two-thirds of the rear tube plate, and all the inside of the tubes come in contact with the products of combustion, and form the *heating surface:* all the heating surface is below the water line.

The boiler setting, shown by Plates II. and III., is brick laid in cement or mortar; all parts that are directly exposed to the fire are lined with fire brick. The walls are laid with confined air spaces to reduce transmission of heat. The *boiler front* is commonly made of cast iron, and has *fire doors* leading to the furnace, and *ash-pit doors* opening from the *ash pit,* or space below the grate; there are also large doors giving access to the tubes, through the *smoke box* at the front end of the boiler. The furnace is formed by the side walls, the bridge, and the lower part of the boiler front, which latter should be lined with fire brick above the grate. Doors through the rear wall give access to the space back of the bridge. The side walls are tied together by through rods and *buck staves.* The top of the boiler is protected by a brick arch or, more commonly, it is covered with non-conducting material.

Two-Flue Boiler. — The cylindrical flue boiler differs from the tubular boiler mainly in replacing the fire tubes by two or more large flues. To provide sufficient heating surface the length of the boiler is greater than that of a tubular boiler of the same size. A two-flue boiler is shown by Fig. 2.

As compared with the tubular boiler it is simpler and gives better access to the interior for cleaning, — advantages which recommend it where water is bad, even though it shows poor coal economy.

Plain Cylindrical Boiler. — In places where fuel is very cheap, especially where it is a waste product, as at saw mills, the plain cylindrical boiler is frequently used. Its external appearance is similar to that of the two-flue boiler (Fig. 2), except that there are no flues and the ends are commonly hemispherical or else curved to a radius equal to the diameter of the shell. Such plain cylindrical boilers are also employed to utilize the waste gases from blast furnaces. They are commonly 30 to 42 inches in diameter and from 20 to 40 feet long. They have been made 70 feet long. With such extreme lengths special care must be taken to ensure equal distribution of the weight over the supports and to provide for expansion.

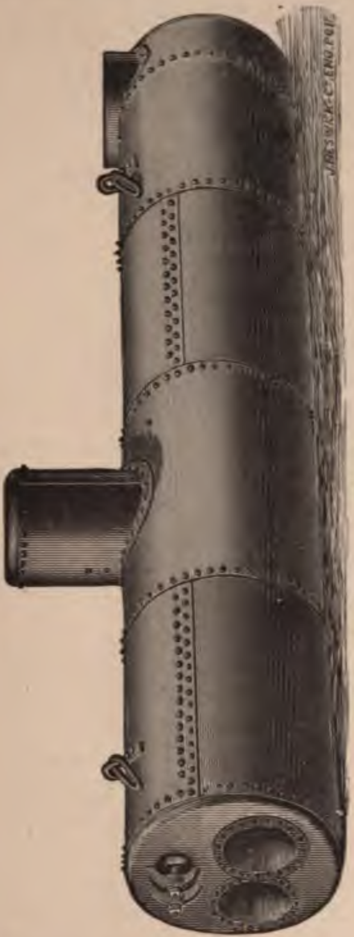

Fig. 2.

Lancashire Boiler. — This boiler, shown by Figs. 3 and 4, is a two-flue shell boiler with furnaces in the tubes; it is therefore an internally fired boiler, in which it differs from the two preceding types, which are externally fired. The chief difficulty in the design of these boilers is to provide sufficiently longer furnaces without making the external shell too large. As compared with the cylindrical tubular boiler, this boiler will be sure to have long, narrow grates, with a shallow ash pit and a low furnace crown: the boiler also appears to be deficient in heating surface. In compensation, radiation and loss of heat from the furnace are almost entirely done away with, and the thick outside shell, with its riveted joints, is not exposed to the fire, as with the tubular boiler. The flues are made in short sections riveted together at the ends, thus forming a series of stiffening rings that add very much to the strength of the flues against collapsing. Conical through tubes, vertical or inclined, give increased heating surface, break up the currents of the hot gases, improve the circulation of the water, and strengthen the flues. These tubes are small enough at the lower end to pass through the hole cut in the flue for the upper end, and thus are readily put in or taken out for repairs.

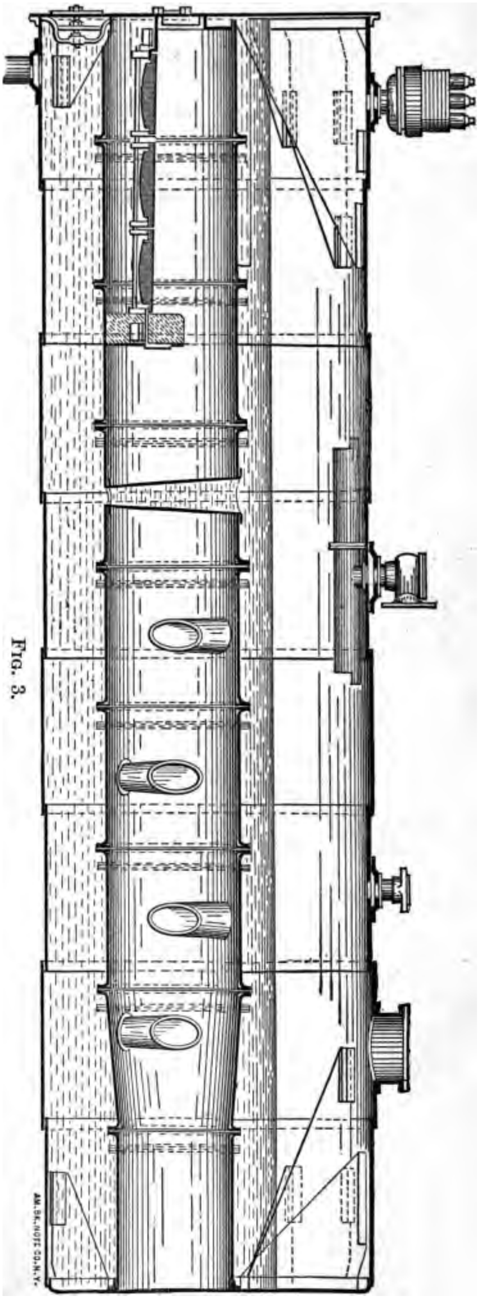

FIG. 3.

The flat plates at the ends of the shell are stayed by *gusset stays* or triangular, flat plates to the shell of the boiler. The boiler is provided with a manhole near the back end, and a safety valve near the front end. Steam is taken through a horizontal dry pipe, perforated on the top.

Galloway Boiler. — This boiler has two furnace flues at the front end, like the Lancashire boiler. Beyond the furnace, the two flues merge into one broad flue, having the upper and lower surfaces stayed by numerous conical through tubes, like those shown in Fig. 3, for the Lancashire boiler.

Cornish Boiler. — This boiler was developed in conjunction with the Cornish engine, and both boiler and engine long had a reputation for high efficiency. It differed from the Lancashire boiler in that it had but one flue; it formerly did not have cross tubes. The one furnace of the Cornish boiler, with a given diameter of shell, can have better proportions than the two furnaces of the Lancashire boiler, but there is even greater difficulty to get sufficient grate area and heating surface. The high economy shown by these boilers when used with the Cornish pumping engine was due to the slow rate of combustion and to the skill and care of the attendant, who was usually the engineer for the pump, and who was stimulated by a system of competition and awards maintained by the mine owners in that district of the country. Figure 5 gives a cross-section of a Cornish boiler.

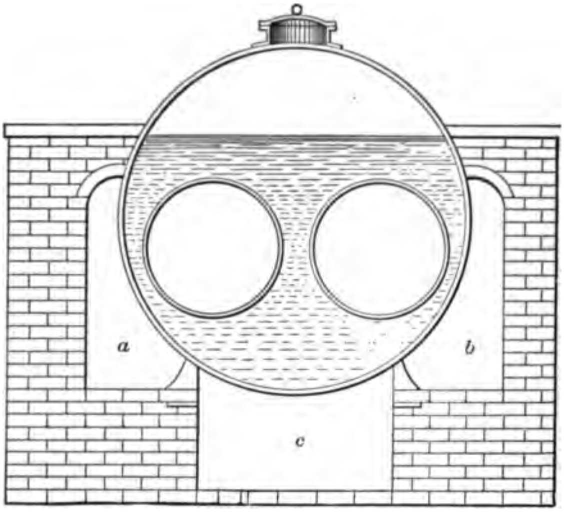

FIG. 4.

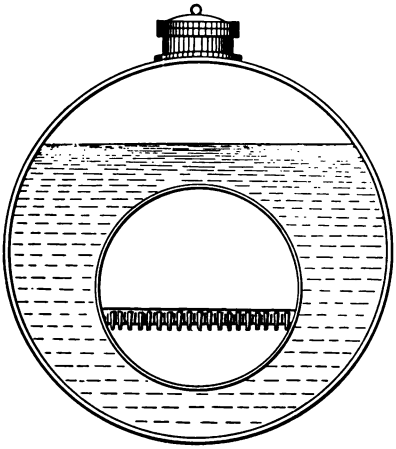

FIG. 5.

Vertical Boilers. — Vertical boilers, like that shown by Fig. 6, are commonly used for small powers; large vertical boilers are used to some extent, and when properly designed and constructed they work satisfactorily and with good economy.

The furnace of a vertical boiler is in a cylindrical fire box in the lower end; a fire door through the double shell gives access to the fire, and an ash pit is provided below the grate; in the figure the ash pit is a casting that forms the base of the boiler. When the fire box is of any considerable size, it is stayed to the outer shell by screw-stay bolts. This form of boiler is powerful for its size and weight, requires no foundation, and is well adapted for hoisting, pile driving, and various other kinds of work requiring a compact, portable boiler; it is wasteful of fuel when made as shown in Fig. 6. Longer vertical boilers are provided with numerous small tubes of considerable length; they may be 16 to 20 feet long and two inches in diameter. Since the tubes extend through the steam space, vertical boilers supply superheated steam. If the boilers are forced and if the tubes are short, they are liable to be burned at the upper ends. To avoid this trouble, a sunken tube plate is used at the upper end of the boiler in order that the water line may be carried above the ends of the tubes. Figure 7 shows a form of upright boiler, with a sunken tube plate, that was at one time commonly used for steam fire engines.

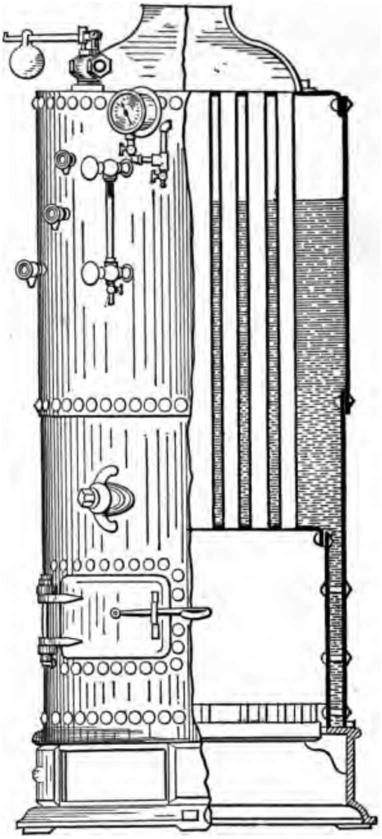

FIG. 6.

At the present time, fire engines have commonly a combination of fire tubes and water tubes. Figures 8 and 9 show the vertical and horizontal sections of the Clapp boiler, built by the American Fire Engine Co. In addition to the fire tubes extending from the lower to the upper tube sheets, there are coiled seamless copper tubes in the combustion chamber over the fire, each of which takes water from the fire-box wall and delivers the heated water, or mixture of water and steam, through the crown sheet. The crown sheet is commonly covered with water to the depth of a few inches, but is protected from the fire by the coiled fire tubes even should it become dry. Though the coiled

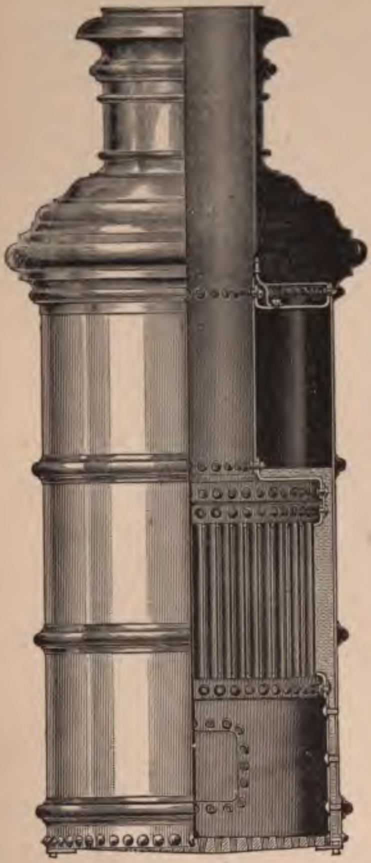

FIG. 7.

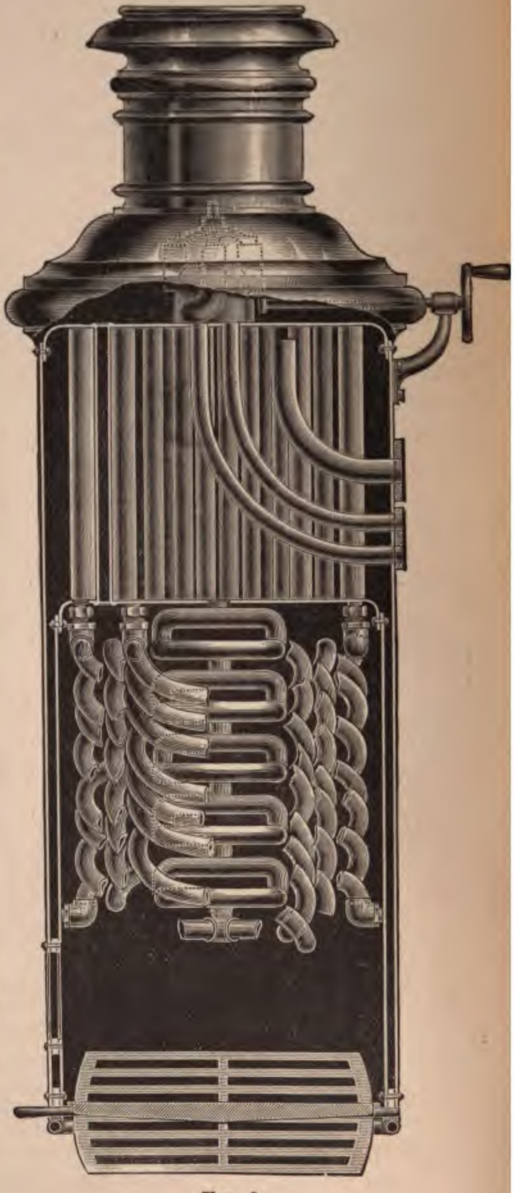

FIG. 9.

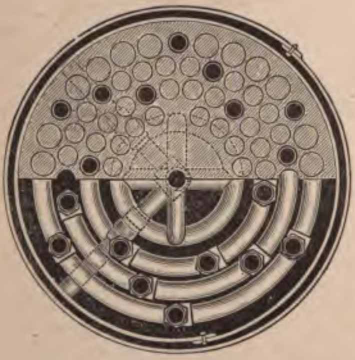

FIG. 8.

tubes are rolled in a helix of small diameter, there is a space inside which would act as a flue and concentrate the draught if it were not occupied. This space is filled by a fire deflector made of several sections; each one consists of a horizontal annular tube with a horizontal diametrical tube, and an inverted vertical U tube connecting with the next higher section. The series of deflectors is supplied with water from the fire-box wall and delivers water and steam through an opening in the crown sheet. Figure 10 shows one section of the deflector.

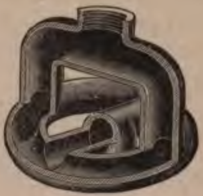

FIG. 10.

Marine Boilers. — A single-ended, two-furnace Scotch boiler is shown in perspective by Fig. 11, and in section and elevation by Plate IV.

The arrangement of the furnaces in the flues is similar to that shown by Figs. 3 and 4 for the Lancashire boiler. The furnace flue leads into a combustion chamber, from which the products of combustion pass through tubes to the uptake at the front end of the boiler.

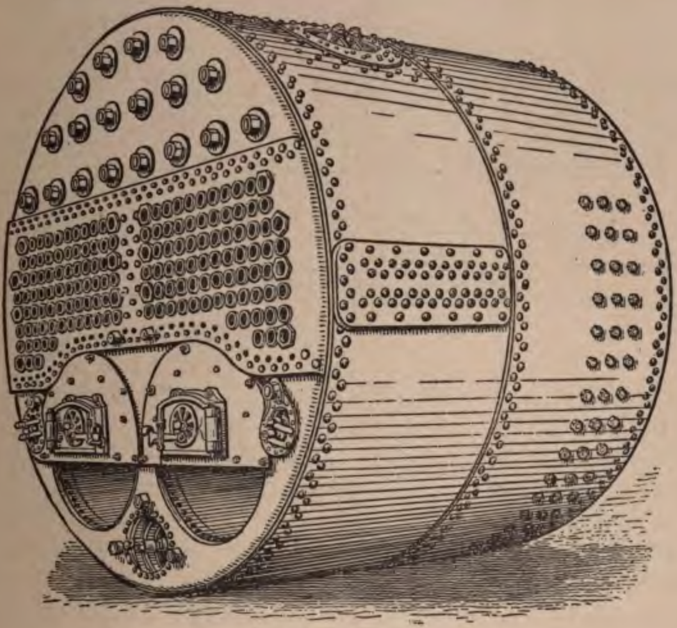

FIG. 11.

Large marine boilers have three or four furnaces. A three-furnace boiler commonly has three combustion chambers, while a four-furnace boiler may have two, into each of which two furnaces lead. Double-ended boilers have furnaces at each end, and resemble two single-ended boilers placed back to back. A double-ended boiler is lighter, cheaper, and occupies less space than two single-ended boilers having the same capacity. In the best practice there are two distinct sets of combustion chambers for the two sets of furnaces; in order to still further lighten double-ended boilers, common combustion chambers for corresponding furnaces at the two ends have been used, but the results have not been satisfactory.

Gunboat Boilers. — In some of the smaller naval vessels, where there is not sufficient head room for a Scotch boiler, the form of boiler shown by Fig. 12 has been used.

Locomotive Boilers. — The locomotive boiler, as shown by Plate V., consists essentially of a rectangular fire box with a grate at the bottom, and a cylindrical barrel, through which numerous tubes pass to the smoke box, which forms a continuation of the barrel and from which the products of combustion pass up the smoke stack. A very rapid rate of combustion of fuel on the grate is obtained by throwing the exhaust steam through the smoke box into the smoke stack, in such a way as to produce a steam blast. Figure 13 gives the external appearance of a boiler of the locomotive type arranged for stationary work. Such a boiler is known as a portable or semi-portable boiler, since it can easily be moved from place to place and requires little or no setting or foundation.

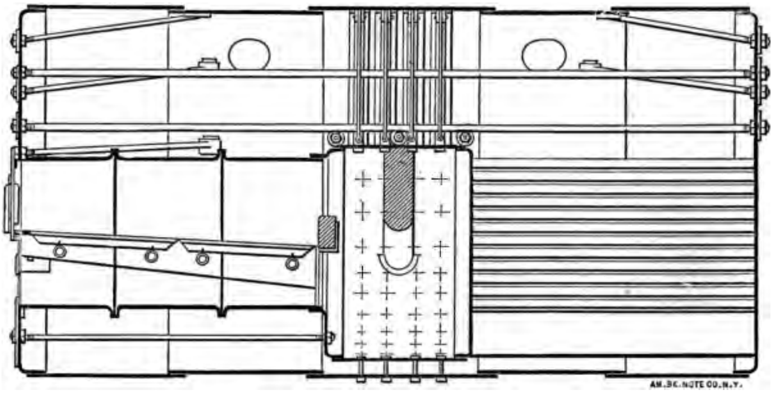

FIG. 12.

Babcock and Wilcox Boiler. — This boiler, which is shown by Figs. 14 and 15, is a water-tube boiler having a cylindrical shell to furnish steam space, and in which is the free-water surface for the disengagement of steam. The tubes are expanded into vertical headers at each end; the front-end headers open into a cross-connection in communication with the cylindrical shell, while the back-end headers are connected with a similar cross-connection by slightly inclined pipes. *The tubes in* each section are staggered so that the tubes taken as

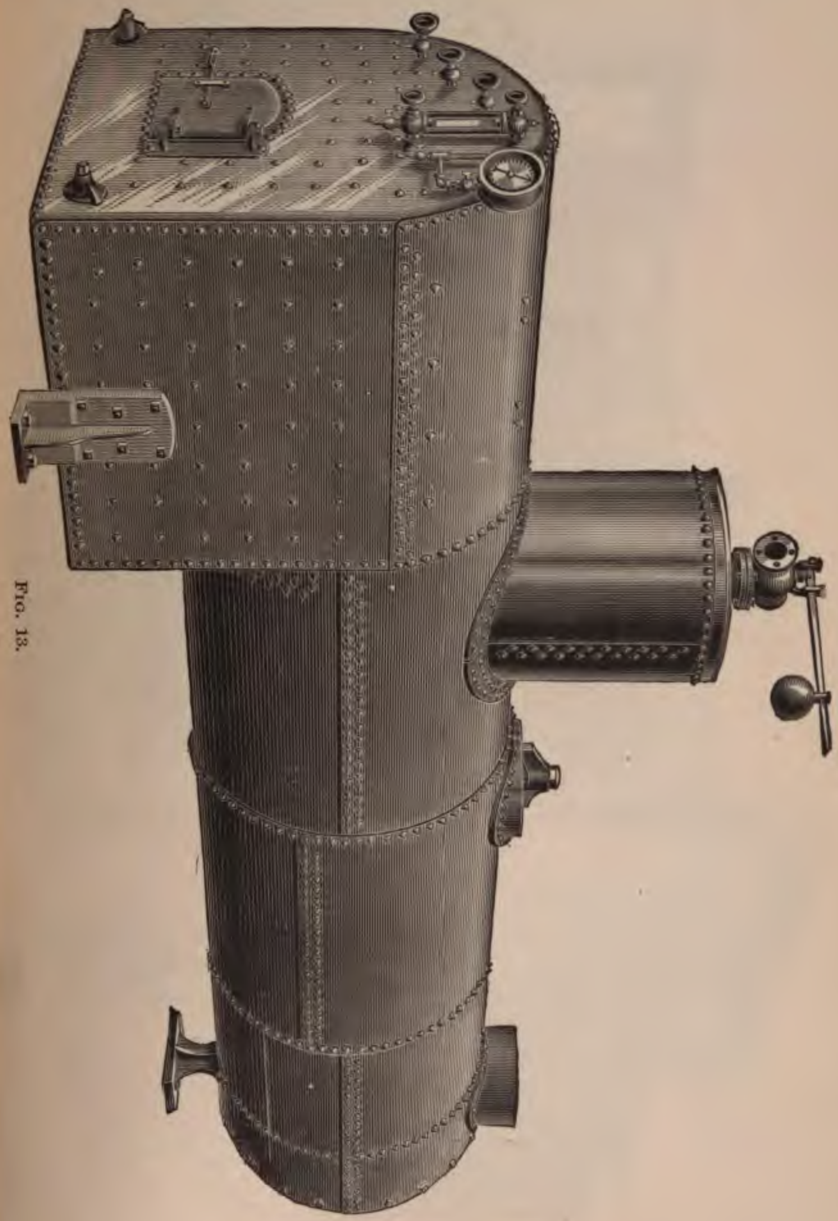

Fig. 13.

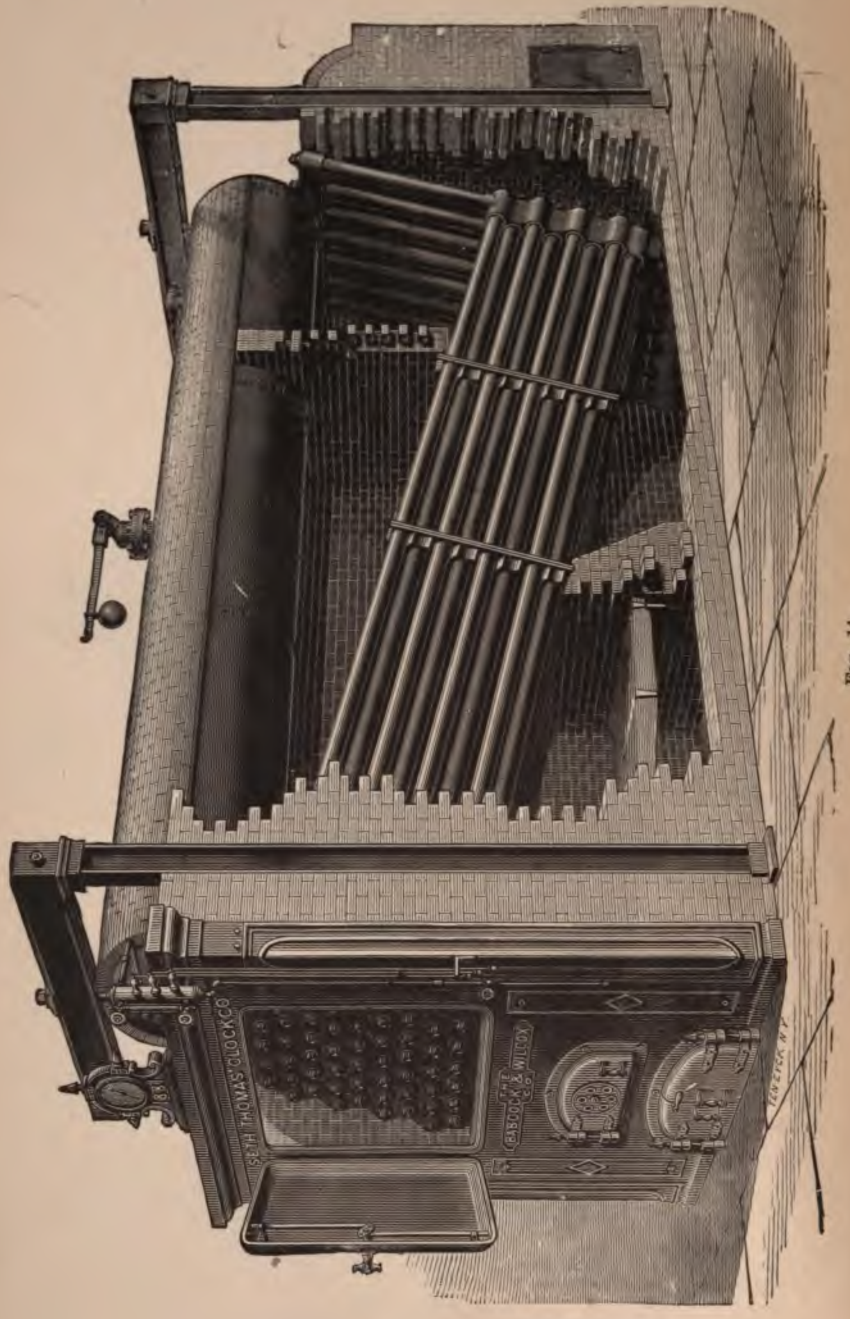

Fig. 14.

a whole are in horizontal rows, but not in vertical rows — an arrangement that gives a better spreading of the products of combustion among the tubes. At each end of each tube are hand holes that give access to the inside of the tube when it needs cleaning or scaling. By the aid of a brick bridge wall at the end of the furnace and a continuation of this wall formed of special tiles, through the tubes, together with a hanging bridge wall similarly continued through the tubes, the products of combustion pass over the tubes three times on the way to the uptake at the back end of the boiler. The lower half of the cylindrical shell serves as heating surface, but it is at such a height above the fire, and is so shielded by the water tubes, that it is not liable to be overheated. The boiler is hung from cross-girders front and back, which in turn are supported on iron columns, and the brick setting is only a screen to retain the heat.

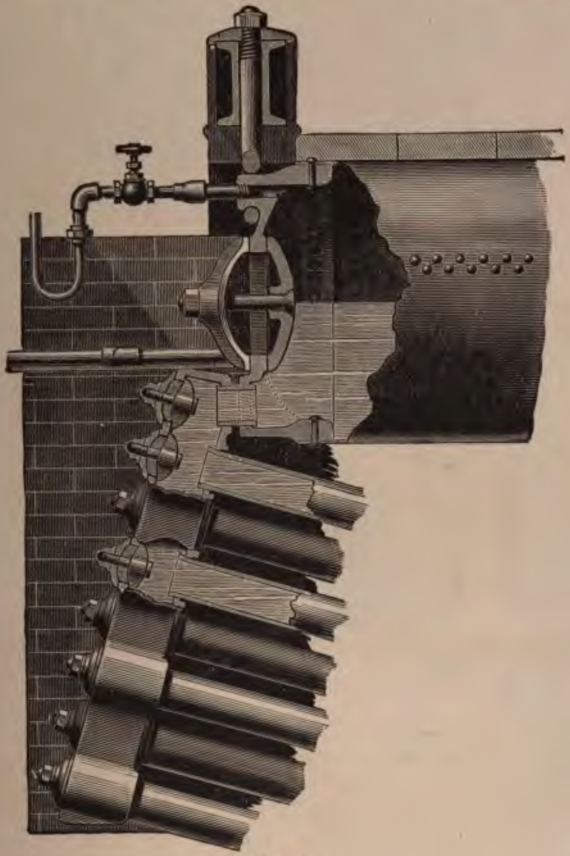

FIG. 15.

The circulation of the water in the boiler is down from the shell at the rear to the water tubes, forward and upward through the tubes, in which course it is partially vaporized and consequently has a less average density, then up into the shell, at the front where the steam and water separate; the water in the shell flows continually from the front to the rear to supply the current through the tubes.

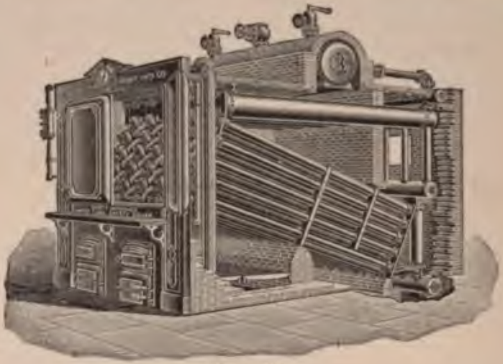

FIG. 16.

The Root Boiler. — The general appearance of the Root boiler is shown by Fig. 16, and details of construction are shown by Fig. 17. Pairs of tubes are first expanded into headers at the end, as shown by 1, Fig. 17; then several pairs are assembled, as shown by 2, to form a vertical *section*, by the aid of bends, of which 3 gives further details. The joints between the bends and headers are made tight by aid of a metallic packing ring shown by 4. The conical bearing on the bend shown by 5 expands the ring into a recess in the header, shown by 6, thus making a steam-tight joint. Each section has a steam drum at the top, as shown in Fig. 16, and at the back end of the steam drum are pipes leading up into the transverse steam drum, and downward into a transverse water pipe at mid-height of the boiler. Near each end of the mid-height water pipe are vertical pipes communicating with the ends of a transverse mud drum, from

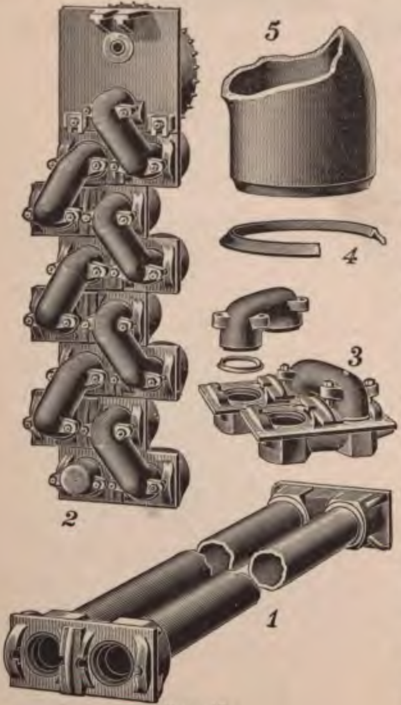

FIG. 17.

which a series of pipes lead to the sections of the boiler. The water circulation is down from the back ends to the mud drums, then ~~back and~~ forth through all the tubes of a section to the front ends of the steam drums, in which the steam and water separate, the steam passing into the transverse steam drum, and the water returning through the back connections to the sections. The products of combustion pass over the tubes ~~three times~~ before escaping to the furnace.

The Thornycroft Boiler. — This boiler, which was specially designed for service in torpedo boats and to work under forced draught, is represented by Figs. 18, 19, and 20.

It consists of two horizontal cylinders placed at the sides of the grate, and one larger cylinder at the top, together with water tubes and circulating pipes. The water tubes are expanded into the cylinders at the top and bottom, and are arranged in longitudinal rows, the tubes in any row being close together, so that the products of combustion cannot pass between them. The products of combustion pass through the spaces between the lower

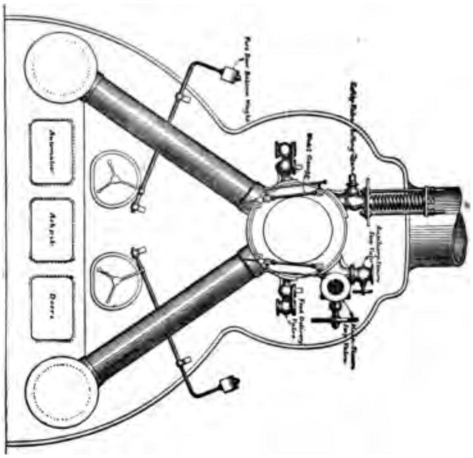

FIG. 18.

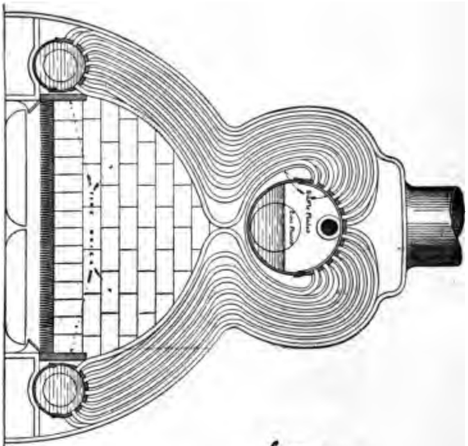

FIG. 19.

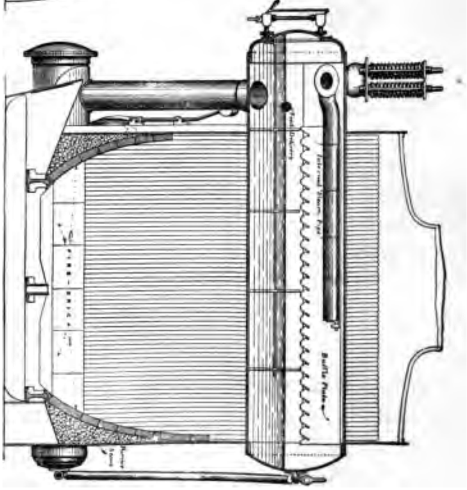

FIG. 20.

ends of the tubes, and thence up among the tubes to the funnel. The circulation of the water is down through the circulating pipes in front of the boiler, into the lower cylinders, thence up through the water tubes, when the rapid vaporization produces a mixture of water and steam that is discharged into the top of the upper cylinder, on a semi-circular shield. The steam separates from the water in the upper cylinder and is withdrawn through a dry pipe under the shield.

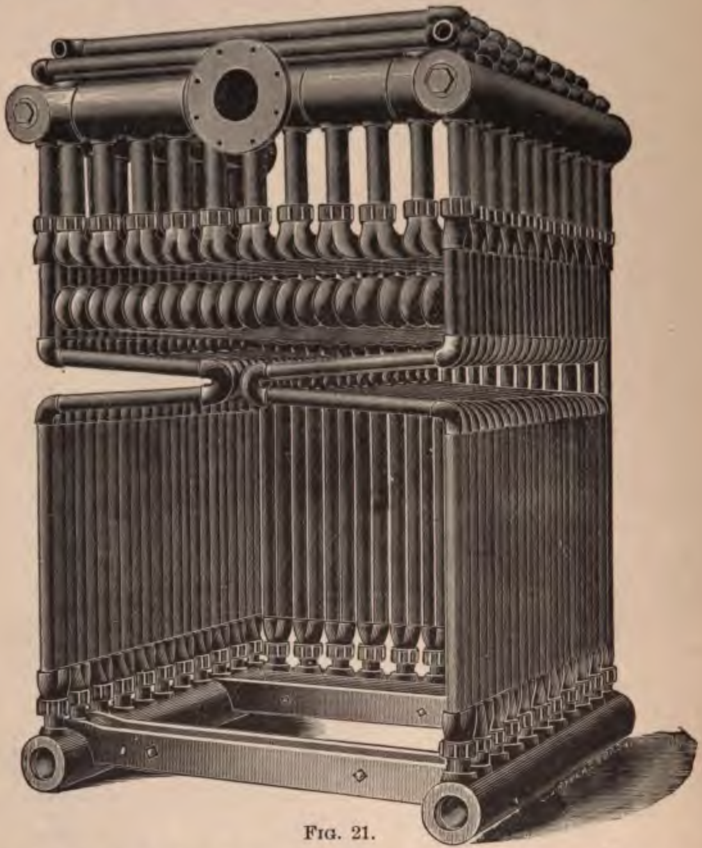

FIG. 21.

The boiler is enclosed in a sheet-iron casing with a lining of non-conducting material.

The Almy Boiler.—This boiler, which is represented by Fig. 21, is made of short lengths of pipe, screwed into return bends and into twin unions. At the bottom is a large tube or pipe forming three sides of a square at the sides and back of the grate. From this water space the tubes lead into a similar structure at the top. The steam and water are discharged into a separator in front of the boiler, from which steam is drawn, while the water separated therefrom, together with the feed water, pass down through circulating pipes to the bottom of the boiler.

CHAPTER II.

FUELS AND COMBUSTION.

THE fuels used for making steam are coal, coke, wood, charcoal, peat, mineral oil, and natural and artificial gas. Various waste and refuse products, such as straw, sawdust, and bagasse are burned to make steam.

All coals appear to be derived from vegetable origin, and they owe their differences to the varying conditions under which they were formed, or to the geological changes which they have undergone.

Anthracite Coal consists almost entirely of carbon and inorganic matters; it contains little if any hydrocarbon. Some varieties, for example certain coals found in Rhode Island, appear to approach graphite in their characteristics, and are burned with difficulty unless mixed with other coals. Good anthracite is hard, compact, and lustrous, and gives a vitreous fracture when broken. It burns with very little flame unless it is moist, and gives a very intense fire, free from smoke. Even when carefully used, it is liable to break up under the influence of the high temperature of the furnace when freshly fired, and the fine pieces may be lost with the ash.

Semi-Anthracite or Semi-Bituminous Coal is intermediate in its properties between anthracite coal and bituminous coal; it contains some hydrocarbon, is less dense than anthracite, it breaks with a lamellar fracture, and it burns readily with a short flame.

Bituminous Coals contain a large and varying per cent of hydrocarbons or bituminous matter. Their physical properties and behavior when burning vary widely, and with all intermediate gradations represented, so that classification is difficult. Three kinds may, however, be distinguished, as follows: —

Dry bituminous coals, which burn freely and with little smoke and without caking.

Caking bituminous coals, which swell up, become pasty and cake together in burning. They are advantageously used for gas making.

Long-flaming bituminous coals, which have a strong tendency to produce smoke; some do and some do not cake while burning.

Coke is made from bituminous and semi-bituminous coal by driving off the hydrocarbons by heat. Coke made as a bi-product in gas retorts is weak and friable and has little value for making steam. Coke made in coking ovens by partial combustion of the coal which is coked, is of a dark gray color, porous, hard, and brittle. It has a metallic lustre and gives out a slight ringing sound when struck. Sulphur in the coal may be burned out in coking, if the coal is moist or if steam is supplied during coking, so that coke may be comparatively free from this noxious element even when made from a poor coal. Coke burns without flame and makes a fierce fire when forced.

Lignite, or brown coal, is of more recent geological formation than coal, and is in a manner intermediate between coal and peat. It frequently contains much moisture and mineral matter. It is used where good coal is difficult to get, and while the better varieties form a useful fuel, the poorer qualities have little value.

Peat, or turf, is obtained from bogs. It consists of slightly decayed roots of the swamp vegetation mingled with more or less earthy matter. For domestic use it is cut and dried in the air. It is little used for making steam, though when pulverized, dried, and compressed it makes a useful artificial fuel.

Wood is used for making steam either in remote places where coal is hard to get and timber is plenty, or where sawdust or other refuse wood is produced in quantity in manufacturing operations. Wood is also used for kindling coal fires. One cord of hard wood is equivalent to one ton of anthracite coal; one cord of yellow pine is equal to half a ton of coal; other soft woods are, as a rule, of less value for fuel.

Charcoal is made by charring wood; it is but little used for making steam.

Mineral Oil, in the form of crude petroleum or the refuse heavy oil left from the distillation of petroleum, is used for making steam, especially in the neighborhood of the Black Sea oil field, and by steamers carrying oil from those fields. It is customary to throw the oil into the furnace in the form of finely divided spray through special spraying apparatus worked either with compressed air or with superheated steam. The use of superheated steam has its convenience only to recommend it, for it adds to the inert material to be uselessly heated. Special precautions must be taken when petroleum is burned, to avoid flooding the furnace with oil and to prevent explosions of the vapor and burning of the oil in tanks or receptacles.

Gases. — Natural gas, from gas wells, has been used for making steam, usually in a crude and wasteful way. Some attempts have been made to use gas made from poor and smoky coal, in producer furnaces like those used in metallurgical operations; but the gain to be expected is only the suppression of the smoke nuisance, which is rather a social than an economical problem.

Artificial Fuels. — The small waste from coals and charcoals, sawdust, and other fine combustible material which cannot be sold in such shape is sometimes made into cakes or briquettes by mixing it with some adhesive material, and then compressing it. The cohesive materials have been wood or coal-tar, or else clay. Tar is available in limited quantities only, and clay is disadvantageous since it adds to the inert material, of which the fine fuel is liable to have an excess. Artificial fuels have some advantages for special purposes, and can be stored compartly; they are used mostly where good fuel is difficult to get.

Composition of Fuels. — The composition of various American and foreign fuels is given in the following table: another table, which gives also the heat of combustion, is given on page 21.

COMPOSITION OF COALS, PER CENTS.

DESCRIPTION OF COAL.	CARBON.	HYDROGEN.	OXYGEN.	NITROGEN.	SULPHUR.	ASH.	SPECIFIC GRAVITY.
ANTHRACITES,							
French	90.9	1.47	1.53	1.00	0.80	4.3	1.35
Welsh	91.7	3.78	1.30	1.00	0.72	1.5	1.37
Rhode Island	85.0	3.71	2.39	1.00	0.90	7.0	1.42
Pennsylvanian	78.6	2.5	1.7	0.8	0.4	14.8	1.45
SEMI-BITUMINOUS,							
Maryland	80.0	5.0	2.7	1.1	1.2	8.3	1.33
Welsh	88.3	4.7	0.6	1.4	1.8	3.2	. . .
BITUMINOUS,							
Pennsylvanian	75.5	4.93	12.35	1.12	1.10	5.0	1.32
Indiana	69.7	5.10	19.17	1.23	1.30	3.5	1.25
Illinois	61.4	4.87	35.42	1.41	1.20	5.7	1.31
Virginian	57.0	4.96	26.44	1.70	1.50	8.4	1.32
Alabama	53.2	4.81	32.37	1.62	1.30	6.7	1.30
Kentucky	49.1	4.95	41.13	1.70	1.40	7.2	1.28
Cape Breton	67.2	4.26	20.16	1.07	1.21	6.1	1.33
Vancouver's Island	66.9	5.32	8.76	1.02	2.20	15.8	1.28
Lancashire gas-coal	80.1	5.5	8.1	2.1	1.5	2.7	. . .
Boghead cannel	63.1	8.9	7.0	0.2	1.0	19.8	. . .
LIGNITES,							
Californian brown coal	49.7	3.78	30.19	1.0	1.53	13.8	1.32
Australian brown coal	73.2	4.71	12.35	1.11	0.63	8.0	1.27
PETROLEUMS,							
Pennsylvanian, crude	84.9	13.7	1.4	. . .	. . .	. . .	0.886
Caucasian, light	86.3	13.6	0.1	. . .	. . .	. . .	0.884
“ heavy	86.6	12.3	1.1	. . .	. . .	. . .	0.938
Petroleum refuse	87.1	11.7	1.2	. . .	. . .	. . .	0.938

Heat of Combustion. — The number of thermal units developed by the complete combustion of one unit of weight of a fuel is called the heat of combustion. It can be determined by burning the fuel in a properly constructed calorimeter. The most recent and best results are those of Mahler,[1] who burns the fuel in a finely divided state in an atmosphere of oxygen at a high pressure, inside of a wrought-iron receptacle or bomb, which is placed in a calorimeter filled with water, and wrapped with non-conducting material. The corrections for this form of calorimeter are determined by burning in it some substance like naphthaline, for which the heat of combustion is known. The calorimeter and the process of using it are simple and direct, the difficulties being those incident to the accurate measurements of temperatures for which the best physical thermometers and considerable special skill are required. While an engineering expert may, and perhaps ought to, acquire the skill required for determining heat of combustion of fuels, an engineer in general practice will commonly find it advantageous to refer such work to a physicist. It is not too much to say that all crude forms of apparatus for determination of heat of combustion of fuels are useless and misleading. The table on the next page gives the results of the determinations of heat of combustion of several fuels which have been selected from those reported by Mahler.

The heats of combustion of carbon in various forms as determined by Berthelot[2] are: —

Diamond	7859	calories.
Diamond bort	7860.9	"
Graphite	7901.2	"
Amorphous from wood	8137.4	"

The following table gives the heat of combustion of some elements and simple gases: —

Carbon burned to CO_2	8,140	calories;	14,650	B.T.U.
Carbon burned to CO			4,400	"
Hydrogen	34,500	"	62,100	"
Marsh gas, CH_4			23,513	"
Olefiant gas, C_2H_4			21,343	"
Carbon dioxide			4,393	"

The great loss from incomplete combustion to CO instead of CO_2 is evident from the table. The heat of combustion given in the table is calculated as follows: the combustion of one pound of carbon produces $\frac{12 + 16}{12} = 2\frac{1}{3}$ pounds of CO, and develops 4400 B.T.U., so that

$$\frac{14,650 - 4400}{2\frac{1}{3}} = 4393.$$

Calculation of Heat of Combustion. — It is customary to calculate the heat of combustion of a fuel on the assumption that the oxygen in the fuel renders inert its equivalent of hydrogen, and that the

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[1] Bulletin de la Societé d'Encouragement pour l'Industrie national, 1891.

[2] Comptes rendu, 1889.

HEAT OF COMBUSTION OF FUELS.

Kind of Coal.	Analysis.					[illegible]	[illegible]	[illegible]	[illegible]
	Carbon.	Hydrogen.	Oxygen and Nitrogen.	Hygroscopic Water.	Ash.				
Anthracite from Pennsylvania	86.456	1.095	2.100	3.450	5.900	3.00	7484	8250	8430
Anthracite from Mure	86.561	1.367	2.900	4.700	4.400	9.75	7408	[illegible]	[illegible]
Semi-anthracite from Kébao	85.740	2.733	2.731	2.800	5.450	5.20	7828	8329	8331
Semi-anthracite from Commentry	84.928	2.892	5.005	1.775	5.400	3.10	7800	8430	8060
Semi-anthracite from Blanzy	82.746	2.916	6.278	1.700	6.300	6.00	7778	8288	8215
Semi-bituminous from Anzin	88.473	4.130	4.338	1.350	1.700	14.08	8292	8656	8707
Semi-bituminous from Aniche	85.937	4.108	5.240	0.025	4.000	11.93	8420	8834	8088
Bituminous from Anzin	83.754	4.385	5.761	1.100	5.000	21.51	8051	8574	8080
Bituminous from the collieries of Saint-Etienne,	84.546	4.772	5.432	1.250	4.000	20.84	8392	8857	8829
Gas-coal from Béthune	82.418	5.080	7.103	1.200	4.100	30.41	8210	8668	8710
Gas-coal from Commentry	80.182	5.245	8.173	3.000	3.400	30.96	7870	8408	8644
Wigan cannel coal	78.382	5.000	5.058	0.000	10.000	31.64	7761	8768	9011
Lignite from Styria	65.455	4.782	24.303	0.710	4.750	50.34	6284	6646	6610
Lignite from Vaugirard	59.795	4.512	25.799	3.144	6.750	49.95	5536	6076	6270
Norwegian pine, partially dried	47.366	5.581	39.780	6.940	0.333	68.93	4477	4828	4947
Cellulose, $C_{12}H_{10}O_{10}$	44.440	6.170	49.390	6.920	0.750	68.93	4200	4200	4264
Tourbe from Bohemia	53.183	5.542	34.230	6.125	0.920	68.93	5489	5903	5609
Coke, Commentry coal	92.727	0.444	2.029	0.500	4.200	68.93	7065	8001	7957
Coke, semi-bituminous coal	94.582	0.633	1.585	0.500	3.200	68.93	7787	8044	8130
Coke, Pennsylvania anthracite	91.036	0.685	2.146	0.233	5.900	68.93	7528	8036	8078

remainder of the hydrogen and all of the carbon in the fuel is available for producing heat. In the form of an equation this may be written,

$$\text{Heat} = 14{,}650\,C + 62{,}100\,(H - \tfrac{1}{8}O),$$

in which C, H, and O represent the weights of carbon, hydrogen, and oxygen found in one pound of fuel.

Mahler finds that the heats of combustion of fuels, determined by him (part of which are given in the table on page 21), can be represented by the empyrical formula, using French units,

$$\text{Heat} = 8140\,C + 34{,}500\,H - 3000\,(O + N).$$

This last equation with English units becomes

$$\text{Heat} = 14{,}650\,C + 62{,}100\,H - 5400\,(O + N).$$

In these equations the letters C, H, O, and N represent the weights of hydrogen, oxygen, and nitrogen (exclusive of ash and hygroscopic moisture) in one kilogramme or one pound of fuel.

Example. — Find the heat of combustion of Pennsylvanian bituminous coal (see page 19).

By the equation at the top of this page,

$$\text{Heat} = 14{,}650 \times 0.755 + 62{,}100\left(0.0493 - \frac{0.1235}{8}\right) = 13{,}166 \text{ B.T.U.}$$

By Mahler's equation,

$$\begin{aligned}\text{Heat} &= 14{,}650 \times 0.755 + 62{,}100 \times 0.0493 - 5400\,(0.1235 + 0.0112)\\ &= 13{,}394 \text{ B.T.U.}\end{aligned}$$

Air required for Combustion. — Neglecting moisture and carbon dioxide, the atmosphere is composed, by weight, of

Oxygen	0.236
Nitrogen	0.764
	1.000

and by volume of

Oxygen	0.213
Nitrogen	0.787
	1.000

For rough calculations it is customary to consider that the atmosphere is made up of one volume of oxygen and four volumes of nitrogen. This approximation is sufficient for calculations of the air required by fuels and for similar purposes, and will be found very convenient.

For convenience, the properties of the elements of fuels and their products of combustion are assembled in the following table: —

	Symbol or Composition.	Atomic or Molecular Weight.	Specific Volumes.	Specific Heat in Gaseous Condition.	Density or Weight of One Cubic Foot.
Carbon	C	12	. . .	. . .	. . .
Hydrogen	H	1	178.881	3.409	0.005590
Oxygen	O	16	11.2070	0.2175	0.08928
Nitrogen	N	14	12.7561	0.2438	0.07837
Carbon dioxide	CO_2	$12 + 2 \times 16$	8.10324	0.2169	0.12341
Carbon monoxide . . .	CO	$12 + 16$	12.81	0.2450	0.07806
Water	H_2O	$2 + 16$	. . .	0.4805	. . .
Air	. . .	. . .	12.3909	0.2375	0.08071
Ash	. . .	. . .	. . .	0.2	. . .

The air required for combustion of a fuel of known composition may be estimated by aid of the chemical composition of the product of combustion. A few examples will make the process clear.

One pound of carbon burned to CO_2 will require

$$2 \times 16 \div 12 = \tfrac{8}{3} = 2\tfrac{2}{3}$$

pounds of oxygen, and, since air is 0.236 part oxygen, the air required will be

$$2\tfrac{2}{3} \div 0.236 = 11.30 \text{ pounds.}$$

If it be assumed that air is composed of one part of oxygen by volume to four of nitrogen, then it appears that

$$(16 + 4 \times 14) \div 16 = \tfrac{9}{2}$$

pounds of air will be required to furnish one pound of oxygen, and we shall then have, for the air required by one pound of carbon,

$$\tfrac{8}{3} \times \tfrac{9}{2} = 12 \text{ pounds.}$$

This result is often quoted and is easily remembered.

One pound of hydrogen burned to H_2O requires

$$\tfrac{1}{2} \times 16 = 8$$

pounds of oxygen, to furnish which

$$8 \div 0.236 = 33.9$$

pounds of air must be supplied.

The assumption that air is one volume of oxygen to four of nitrogen, gives in this case

$$8 \times \tfrac{9}{2} = 36$$

pounds of air per pound of hydrogen.

The combination of the approximate estimates of the quantity of air required for carbon and hydrogen, with the assumption that is used in estimating the heat of combustion of fuels, namely, that the oxygen in the fuel renders inert its equivalent of hydrogen, and that the remainder of the hydrogen is available for combustion, gives the following equation, —

$$\text{Air} = 12\,C + 36\,(H - \tfrac{1}{8}\,O),$$

for calculating the air required for a given fuel. In this equation C, H, and O represent the weights of carbon, hydrogen, and oxygen in one pound of fuel.

An application of the above formula to Pennsylvanian bituminous coal, for which the analysis appears on page 19, gives

$$\text{Air} = 12 \times 0.755 + 36\left(0.0493 - \frac{0.1235}{8}\right) = 10.3 \text{ pounds.}$$

This result is somewhat larger than would be obtained were the more exact composition of the atmosphere given on page 22 used, together with the assumption that the oxygen renders inert its equivalent of hydrogen; but the method is not sufficiently well grounded to warrant much refinement.

As a further illustration of the method, the following calculation of the air required for one pound of olefiant gas may be interesting. This gas, having the composition C_2H_4, consists of

$$\frac{2 \times 12}{2 \times 12 + 4 \times 1} = \tfrac{6}{7} \text{ carbon,}$$

$$\frac{4 \times 1}{2 \times 12 + 4 \times 1} = \tfrac{1}{7} \text{ hydrogen,}$$

and will require

$$\tfrac{6}{7} \times 11.24 + \tfrac{1}{7} \times 33.9 = 14.5 \text{ pounds of air.}$$

Air for Dilution. — In order to secure complete combustion of coal in the furnace of a boiler, it is necessary to supply an excess of oxygen, or, what amounts to the same thing, an excess of air. This excess varies from one-half the quantity required for combustion to an equal quantity. Thus, roughly, from 18 to 24 pounds of air may be furnished per pound of carbon, and from 54 to 72 pounds of air per pound of hydrogen.

Volume of Air for Combustion. — The specific volume of air at a given pressure and temperature may be calculated from the characteristic equation for perfect gases, which can be conveniently written

$$\frac{pv}{p_0v_0}=\frac{T}{T_0},$$

in which p_0, v_0, and T_0 represent the specific pressure, the specific volume, and the absolute temperature at the freezing-point of water and at normal atmospheric pressure, while p, v, and T represent the corresponding properties under the given conditions.

The normal pressure of the atmosphere may be taken to be 14.7 pounds per square inch, or 29.92 inches of mercury. The variations of the pressure of the atmosphere may frequently be neglected in this work. For a rough estimate, the specific volume of air at 60° F. may be taken to be 13 cubic feet, and then the cubic feet of air required per pound of the elements of fuel will be represented by the following table: —

	Without dilution.	With 50 % dilution.	With 100 % dilution.
Carbon	150	225	300
Hydrogen	500	725	1000

A more careful estimate should be made for each spiral case, as illustrated by the following example: —

Example. — Required the weight and volume of air required for combustion of Pennsylvanian bituminous coal (see page 19) with 50 per cent dilution, the temperature of the atmosphere being 70° F. and the height of the barometer (reduced to 32° F.) being 29 inches.

Assuming that the oxygen in the coal renders inert its equivalent of hydrogen, the coal contains

$$4.93-\frac{12.35}{8}=3.39 \text{ per cent}$$

of available hydrogen, and 75.5 per cent of carbon. The oxygen required for combustion will be

$$2\tfrac{2}{3}\times 0.755+8\times 0.0339=2.284 \text{ pounds.}$$

The air required by the fuel without dilution will be

$$2.284\div 0.236=9.7 \text{ pounds.}$$

With 50 per cent dilution, the air required will be 13.5 pounds per pound of coal.

The specific volume of air at 70° F. and at 29 inches of mercury is

$$12.39\times\frac{530.7}{492.7}\times\frac{29.92}{29}=13.77,$$

so that the volume of air required per pound of coal will be

$$13.8 \times 9.7 = 134 \text{ cubic feet,}$$

and the volume with 50 per cent for dilution will be 200 cubic feet.

Air per Pound of Coal. — The amount of air supplied per pound of coal may be determined either by measuring the air supplied to the furnace or by an analysis of the products of combustion.

For the first method the following arrangement has been used in boiler tests at the Institute of Technology. The ash-pit doors are removed and a sheet-iron mouthpiece is fitted over the opening into the ash-pit. The air for combustion is supplied by a cylindrical sheet-iron conduit leading into this mouthpiece. The area of the conduit should be at best equal to the area of the fire door or fire doors, and its length should be several times its diameter. The velocity of the air in the conduit is measured by an anemometer from which the volume of air is readily calculated, and its weight determined from the temperature and pressure of the atmosphere. The joint between the mouthpiece and the furnace front must be luted to avoid leakage, and leaks or admission of air to the furnace otherwise than through the sheet-iron conduit must be stopped or allowed for. Anemometers, even when tested and rated, are liable to be affected by errors of two per cent or more. They are commonly tested by swinging them on a revolving arm through still air, a method that is proper for small or moderate velocities, but difficult to use, and is vitiated by the action of centrifugal force at high speeds. An ideal way of testing an anemometer would be to find its reading in such a conduit when the weight, and consequently the velocity, of the air per second is known. The weight may be determined by causing the supply of air to flow through a well-rounded orifice, to which calculations by Flügner's equations may be applied. This method for large conduits would involve the use of a very large air compressor, which makes it hardly practicable.

Orsat's Gas Apparatus. — This apparatus, which is well adapted to the analysis of flue gases, determines the proportion by volume of the carbon dioxide, carbon monoxide, and oxygen in the air. The remainder of the flue gases is commonly assumed to be nitrogen, but it includes unburned hydrocarbon, if there be any, and steam or vapor of water. In Fig. 22, *A*, *B*, and *C* are pipettes containing, respectively, solution of caustic potash to absorb carbon dioxide, pyrogallic acid and caustic potash to absorb oxygen, and cuprous chloride in hydrochloric acid to absorb carbon monoxide.

At *W* is a three-way cock to control the admission of gas to the apparatus; at *D* is a graduated burette for measuring the volumes of gas, and at *P* is a pressure bottle connected with *D* by a rubber tube to control the gases to be analyzed. The pressure bottle is commonly filled with water, but glycerine or some other fluid may be used when, in addition to the gases named, a determination of the moisture or steam in the flue gases is made.

The several pipettes *A*, *B*, and *C* are filled to the marks *a*, *b*, and *c* with the proper reagents, by aid of the pressure bottle *P*. With the three-way cock to open to the atmosphere, the pressure bottle *P* is raised till the burette *D* is filled with water to the mark *m*; communication is then made with the flue, and by lowering the pressure bottle the burette is filled with the gas to be analyzed, and two minutes are allowed for the burette to drain. The pressure bottle is now raised till the water in the burette reaches the zero mark and the clamp *k* is closed. The valve *W* is now opened momentarily to the atmosphere to relieve the pressure in the burette. Now open the clamp *k* and bring the level of the water in the pressure bottle to the level of the water in the burette, and take a reading of the volume of the gas to be analyzed; all readings of volume are to be taken in a similar way. Open the cock *a* and force the gas into the pipette *A* by raising the pressure bottle, so that the water in the burette comes to the mark *m*. Allow three minutes for absorption of carbon dioxide by the caustic potash in *A*, and finally bring the reagent to the mark *a* again. In this last operation, brought about by lowering the pressure bottle, care should be taken not to suck the caustic reagent into the stopcock. The gas is again measured in the burette and the diminution of volume is recorded as the volume of carbon dioxide in the given volume of gas. In like manner the gas is passed into the pipette *B*, where the oxygen is absorbed by the pyrogallic acid and caustic potash; but as the absorption is less rapid than was the case with the carbon monoxide, more time must be allowed, and it is advisable to pass the gas back and forth, in and out of the pipette several times. The loss of volume is recorded as the volume of oxygen. Finally, the gas is passed into the pipette *C*, where the carbon monoxide is absorbed by cuprous chloride in hydrochloric acid.

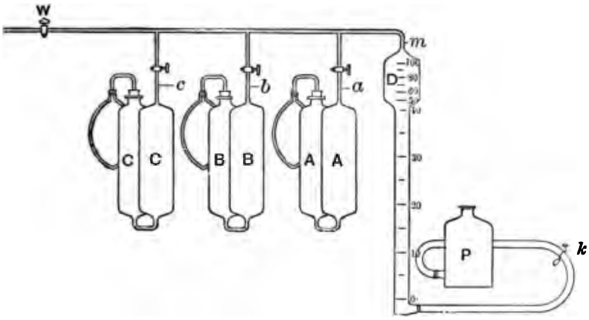

FIG. 22.

The solutions are as follows: —

A. Caustic potash, 1 part; water, 2 parts.
B. Pyrogallic acid, 1 gramme to 25 c.c. caustic potash.
C. Saturated solution of cuprous chloride in hydrochloric acid having a specific gravity of 1.10.

The absorption values per cubic centimetre of the reagents are

A. Caustic potash absorbs 40 c.c. carbon dioxide.
B. Pyrogallate of potassium absorbs 22 c.c. oxygen.
C. Cuprous chloride absorbs 6 c.c. carbon dioxide.

Calculation for Gas Analysis. — The calculation of the amount of air supplied per pound of carbon and per pound of coal, from the known chemical constituents of the flue gases, is best shown by an example.

Example. — Let it be assumed that the analysis of the flue gases resulting from the burning of Pennsylvanian bituminous coal gives by volume, 13 per cent of carbon dioxide, 0.5 per cent of carbon monoxide, and 6 per cent of oxygen. It is convenient to treat the percentages by volume as the number of cubic feet of the several gases in one hundred cubic feet of flue gas. We will thus have: —

Gas.	Volume.	Density.	Weight.
Carbon dioxide	13	0.12341	1.6043
Carbon monoxide	0.5	0.07806	0.03903
Oxygen	6	0.08928	0.53568

Now one pound of carbon dioxide is composed of

$$\frac{2 \times 16}{12 + 2 \times 16} = \frac{8}{11}$$

of a pound of oxygen and $\frac{3}{11}$ of a pound of carbon. And a pound of carbon monoxide is composed of

$$\frac{16}{12 + 16} = \frac{4}{7}$$

of a pound of oxygen and $\frac{3}{7}$ of a pound of carbon. Consequently we have

$$\begin{array}{lrlr}
\frac{8}{11} \times 1.6043 & = 1.1668 & \frac{3}{11} \times 1.6043 & = 0.4375 \\
\frac{4}{7} \times 0.03903 & = 0.0223 & \frac{3}{7} \times 0.03903 & = 0.0167 \\
 & 0.5357 & & \\
\hline
\text{Pounds of oxygen} & 1.7248 & \text{Pounds of carbon} & 0.4542
\end{array}$$

and as air consists of 0.236 part by weight of oxygen, the air per pound of carbon from the gas analysis is

$$\frac{1.7248}{0.4542} \div 0.236 = 16.1 \text{ pounds.}$$

Since the coal in question contains 75.5 per cent of carbon, the air apparently supplied per pound of coal is

$$0.755 \times 16.1 = 12.2 \text{ pounds.}$$

Now, we have hitherto made use of the convention, that the oxygen in the fuel renders inert one-eighth of its weight of hydrogen, and that the remnant is available for combustion. On this supposition there should be added to the air per pound of coal, just formed,

$$36\left(0.0493 - \frac{0.1235}{8}\right) = 1.2 \text{ pounds,}$$

making the total air per pound of coal,

$$12.2 + 1.2 = 13.4 \text{ pounds.}$$

Hypothetical Temperature of Combustion. — A hypothetical maximum temperature of combustion of a fuel may be calculated from its heat of combustion and the weights and specific heats of its products of combustion; the temperatures thus found are commonly much greater than those actually realized in the furnace of a boiler.

For example, the combustion of one pound of carbon produces

$$\frac{12 + 2 \times 16}{12} = 3\tfrac{2}{3} \text{ pounds}$$

of carbon dioxide and (page 23) requires 11.24 pounds of air, so that the weight of the products of combustion is 12.24 pounds, of which

$$12.24 - 3\tfrac{2}{3} = 8.57 \text{ pounds}$$

are nitrogen. The number of thermal units required to raise the temperature of the products of combustion one degree F. is found as follows: —

	Weight.		Specific heat.	
Carbon dioxide	$3\frac{2}{3}$	×	0.2169	= 0.7953
Nitrogen	8.57	×	0.2438	= 2.0894
Thermal units per degree F.				2.8847

If it be assumed that the specific heats are constant at all temperatures, and that all the heat can be applied to raising the temperature, then the elevation of temperature will appear to be

$$14,650 \div 2.8847 = 5078° \text{ F.}$$

The hypothetical temperature of combustion of a coal may be calculated on the assumption that the carbon and hydrogen appear in the products of combustion as carbon dioxide and superheated steam. For this purpose all the hydrogen in the coal is taken into account, for it will make no difference whether the steam results from the combustion of the available hydrogen or whether it results from the breaking up of the compounds of the coal.

Example. — Find the hypothetical elevation of temperature resulting from the combustion of Pennsylvanian bituminous coal, with 50 per cent dilution. The example on page 22 gives for the heat of combustion 13,166 B.T.U., on the assumption that the oxygen in the fuel renders inert one-eighth of its weight of hydrogen, which assumption was used also for the example on page 25. It also appears from this latter example that the coal in question requires 2.284 pounds of oxygen or 9.7 pounds of air for combustion, and 4.9 pounds are allowed for dilution. The nitrogen in the fuel added to that brought in by the air required for combustion will give for the total nitrogen

$$\begin{array}{r} 9.7 - 2.126 = 7.416 \\ .0112 \\ \hline 7.4272 \text{ pounds.} \end{array}$$

The products of combustion include also

$$\begin{aligned} 3\tfrac{2}{3} \times 0.755 &= 2.768 \text{ pounds } CO_2 \\ 9 \times 0.0493 &= 0.4437 \text{ pounds } H_2O. \end{aligned}$$

	Weight.		Specific heat.	
CO_2	2.768	×	0.2169	= 0.6004
H_2O	0.4437	×	0.4805	= 0.2132
N	7.4272	×	0.2438	= 1.8108
Air	4.9	×	0.2375	= 1.1638
Ash	0.05	×	0.2	= 0.10
				3.8882

The elevation of temperature appears to be

$$13,166 \div 3.8882 = 3386 \text{ F.}$$

The sulphur in the coal has been neglected as well as the heat of vaporization of the water, to avoid further complication. If it be assumed that the temperature of the air and the coal before combustion was 70° F., then the temperature of the fire would appear to be

$$3386 + 70 = 3456° \text{ F.}$$

The temperature of the furnace never reaches the figures resulting from the calculation just made; it seldom or never reaches the hypothetical temperature. In the first place, dissociation is liable to appear even at lower temperature than that given for Pennsylvanian coal; and in the second place, the glowing fuel and the incandescent flames radiate heat energetically to the boiler and furnace, so that it is doubtful if the temperature of dissociation is ever reached in the furnace of a boiler. A test on Babcock and Wilcox boilers gives a temperature of about 1100° F. above the fire and 400° F. in the chimney. The temperature of the gases in the uptake of a boiler depends on the type of the boiler, the way in which it is run, and on the steam pressure in the boiler; it may vary from 350° F. to 600° F. or more.

A test[1] on a boiler of the locomotive type, at the Boston Main Drainage Station, gave for the temperature of the gases escaping from the boiler 439° F., while the steam in the boiler was about 337° F. The gases were afterwards reduced to 194° F. by passing them through a feed-water heater. This boiler was designed for and gave a high efficiency, and the results obtained may be considered to represent first-rate practice.

Volume of the Products of Combustion. — The volume of the products of combustion can be calculated for any given temperature and pressure. Here, again, the method is most easily explained by an example.

Example. — Find the volume of the products of combustion of Pennsylvanian bituminous coal, with 50 per cent dilution, at 450° F. and at 28.5 inches of mercury.

Using the weights of the products of combustion already calculated on page 30, we have the following calculation for the volume of the gases, other than steam, at freezing point and at the pressure of the atmosphere.

	Weight.	Specific volume.	
CO_2	2.768	× 8.10324	= 22.43
N	7.4272	× 12.7561	= 94.74
Air	4.5	× 12.3909	= 55.76
Volume at 32° F. and at 29.92 inches high . .			172.93

The volume at 450° F. and at 28.5 inches of mercury will be

$$172.93 \times \frac{29.92}{28.5} \times \frac{910.7}{492.7} = 336 \text{ cubic feet.}$$

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[1] Thermodynamics of the Steam Engine, page 299.

The specific volume of superheated steam at the given temperature may be calculated by the characteristic equation

$$pv = 93.5\,T - 971\,p^{\frac{1}{4}},$$

$$p = 2116.32 \times \frac{28.5}{29.92} = 2016 \text{ pounds per square foot,}$$

so that for 0.4437 of a pound of steam (see page 30), the volume is

$$0.4437\,v = 0.4437 \frac{93.5 \times 910.7 - 971 \times \overline{2016}^{\frac{1}{4}}}{2016}$$

$$= 17.3 \text{ cubic feet,}$$

and the total volume of the products of combustion is

$$336 + 17 = 351 \text{ cubic feet.}$$

This result may be compared with the volume of the air supplied to the coal per pound, shown on page 26.

CHAPTER III.

CORROSION AND INCRUSTATION.

THE water supplied to a boiler for forming steam may corrode the iron of the boiler, or it may deposit material that can form a scale or incrustation; both actions may go on at the same time.

Pure water, free from air and carbon dioxide, has little or no solvent action on iron, even though some other metal, such as copper, which may with the iron form the elements of a galvanic couple, be present. On the other hand, iron will not rust if placed in an atmosphere of dry air or dry carbon dioxide. All natural water, rain water, water from wells, rivers, lakes, or the sea contains air in solution, and carbon dioxide is not infrequently found in such waters. Iron is rapidly acted upon by water containing air or carbon dioxide, and, on the other hand, iron rusts rapidly in air or carbon dioxide when moisture is present. Again, distilled water, as from the surface condenser of a marine engine containing more or less oil, or the substances resulting from the action of steam on oil, causes corrosion in boilers that are free from scale. To avoid rusting of boilers when not in use they ought to be either quite dry inside, or they ought to be entirely filled with water; preferably water that has been freed from air by boiling. In the American navy it has been the custom to dry out boilers and paint them inside with mineral oil preparatory to laying them up. In the English navy the boilers are dried out, a pan of glowing charcoal is placed in the boiler to consume the oxygen of the air, and quicklime is introduced to absorb moisture.

Mineral Impurities. — The impurities found in water supplied to land boilers are commonly carbonate of lime and sulphate of lime, with more or less organic matter, and sometimes sand or clay held in suspension. The following table gives the number of grains of various mineral substances held in solution in water from several sources: —

Water supplied to land boilers is either hard or soft; the first contains appreciable quantities of lime, and the other usually contains little solid matter of any sort. The first three examples in the table on the next page may be taken as typical soft waters, and all the others, except the last two, as typical hard waters. While there is considerable difference in the amounts and the composition of the solids in solution in the several examples of hard water, it will be seen that they are all characterized by a considerable amount of calcium and magnesium car-

MINERAL MATTER IN SOLUTION. GRAINS PER U. S. GALLON.

	Charles River.	Long Pond.	Schuylkill River.	Lake Michigan.	Mississippi River.	Missouri River at Council Bluffs.	Mississippi River at Keokuk.	Riverside, Ill. Well.	Downer's Grove, Ill. Well very bad.	Rockford, Ill. Artesian Well.	Dead Sea Water.
Silica (SiO_2)	. . .	. . .	0.0800	0.306	0.863	1.522	1.190	0.484	0.741	0.624	. . .
Calcium carbonate ($CaCO_3$)	0.1610	. . .	1.8720	4.461	6.870	8.847	4.673	5.237	17.091	8.141	. . .
Calcium sulphate ($CaSO_4$)	0.2624	. . .	. . .	0.309	0.484	2.251	. . .	0.776	14.037	. . .	. . .
Calcium chloride ($CaCl_2$)	0.0420	0.0308	. . .	. . .	. . .	. . .	. . .	. . .	. . .	. . .	29.220
Magnesium carbonate ($MgCO_3$)	0.0399	. . .	0.3510	2.200	4.006	1.866	0.857	4.023	. . .	7.336	. . .
Magnesium sulphate ($MgSO_4$)	. . .	0.1020	0.0570	. . .	0.338	3.505	. . .	. . .	25.422	. . .	. . .
Magnesium chloride ($MgCl_2$)	. . .	0.0764	. . .	. . .	. . .	. . .	. . .	. . .	. . .	. . .	50.950
Magnesium bromide	. . .	. . .	. . .	. . .	. . .	. . .	. . .	. . .	. . .	. . .	7.950
Sodium carbonate (Na_2CO_3)	. . .	. . .	. . .	. . .	. . .	. . .	2.129	. . .	. . .	. . .	. . .
Sodium sulphate (Na_2SO_4)	0.3816	. . .	. . .	. . .	. . .	. . .	. . .	. . .	. . .	0.554	. . .
Sodium chloride ($NaCl$)	0.1547	0.0323	0.1470	0.225	. . .	. . .	0.100	. . .	. . .	0.362	78.650
Potassium carbonate (K_2CO_3)	. . .	. . .	. . .	. . .	. . .	. . .	. . .	. . .	. . .	. . .	. . .
Potassium sulphate (K_2SO_4)	. . .	. . .	. . .	0.283	. . .	. . .	0.430	. . .	. . .	0.525	. . .
Potassium chloride (KCl)	. . .	0.0380	. . .	. . .	. . .	. . .	0.489	. . .	. . .	. . .	. . .
Ferrous carbonate ($FeCO_3$)	. . .	. . .	. . .	0.029	. . .	. . .	2.682	. . .	. . .	. . .	. . .
Alumina (with ferric oxide)	. . .	0.0800	. . .	. . .	0.233	0.233	. . .	0.146	0.192	0.087	. . .
Organic matter, etc.	0.5291	0.5295	1.6436	. . .	. . .	. . .	. . .	. . .	. . .	. . .	. . .
Suspended mineral matter	. . .	. . .	. . .	. . .	. . .	. . .	1.802	. . .	. . .	. . .	. . .
Suspended organic matter	. . .	. . .	. . .	. . .	. . .	. . .	2.455	. . .	. . .	. . .	. . .

bonates, and (with the exception of Nos. 6 and 9) accompanied by a comparatively small amount of calcium and magnesium sulphates. It will be noticed that Missouri River water is distinctly worse than Mississippi River water, not only in that it contains more of the carbonates, but because it contains a considerable quantity of sulphates. No. 9, from a well at Downer's Grove on the C. B. and Q. R. R., a few miles from Chicago, has been selected as an example of a very bad hard water, especially as it contains so much sulphate. The reason for considering the sulphates of lime and magnesia so deleterious will appear a little later. Note will be made that the water from the Mississippi River at two different places, and presumably at different seasons of the year, vary considerably, especially in the amount of matter held in suspension.

In some places in the western parts of the United States the only available waters for making steam are strongly impregnated with alkalies and borax. Such waters have so deleterious an action on boilers that the advisability of using a surface condenser, as at sea, the distillation of water by a multiple effect evaporator, or the introduction of a supply of good water even from remote places, is worthy of consideration. If the use of such water cannot be avoided, a competent chemist should be consulted to suggest methods for ameliorating the bad effects so far as possible. As each case is liable to require special treatment, no further discussion appears profitable in this place.

The carbonates of lime and magnesia are held in solution in water by an excess of carbon dioxide, and are completely precipitated by boiling. They are thrown down from water supplied to a boiler, in the form of a white or grayish mud, provided there are not other impurities that cement them together and form a hard scale. The customary and sufficient method of treating boilers supplied with water containing carbonates of lime and magnesia is to let the boiler, while full, cool down, and then run out the water and thoroughly wash out the boiler with a strong stream from a hose. If the water is blown out under steam pressure, the deposits are hardened and are removed with difficulty. While pure carbonates are easily treated as just described, the presence of other impurities, such as oil or organic matter, or of sulphate of lime, is likely to make the deposits hard and adhering.

Sulphate of lime is much more soluble in cold than in hot water, and is entirely thrown down from water at a temperature of 280° F., corresponding to 35 pounds pressure of steam above the atmosphere. It forms a hard and adhering scale, and even in comparatively small quantities has a bad effect on scales and deposits composed of carbonates, as has already been suggested. The bad effect of deposits from water containing calcium sulphate is much ameliorated by introducing soda crystals or soda ash into the boiler with the feed water. The result is to give a deposit of calcium carbonate in the form of a fine white powder, which must be washed or swept out, and sodium sulphate in solution, which must be blown out from time to time.

If the mineral matters in the water are known from a chemical analysis, the quantity of soda crystals to be used may be calculated as follows: —

Example. — Find the weight of soda crystals required per day for a boiler supplied with 1000 gallons of water per day from the well at Downer's Grove.

From the table on page 34 it appears that each gallon of the water contains 14.037 grains of $CaSO_4$ and 25.422 grains of $MgSO_4$. The formula for soda crystals being $NA_2CO_3 + 10\,H_2O$, the reactions, neglecting the water of crystallization, will be

$$CaSO_4 + Na_2CO_3 = CaCO_3 + Na_2SO_4$$
$$MgSO_4 + Na_2CO_3 = MgCO_3 + Na_2SO_4.$$

If x_1 is the grains of soda crystals to act on the calcium, we have

$$CaSO_4 : Na_2CO_3 + 10\,H_2O = 14.037 : x$$
$$40 + 32 + 4 \times 16 : 2 \times 23 + 12 + 3 \times 16 + 10\,(2 + 16)$$
$$= 14.037 : x.$$
$$\therefore\ x_1 = 29.52 \text{ grains.}$$

The magnesium sulphate which is soluble is also changed into the carbonate and thrown down as a white precipitate, adding to the deposit. The number of grains of soda crystals required for this reaction is found as follows: —

$$MgSO_4 : Na_2CO_3 + 10\,H_2O = 25.422 : x_2$$
$$24 + 32 + 4 \times 16 : 2 \times 23 + 12 + 3 \times 16 + 10\,(2 + 16)$$
$$= 25.422 : x_2.$$
$$\therefore\ x_2 = 60.59 \text{ grains.}$$

The total weight of soda crystals per gallon is therefore

$$29.52 + 60.59 = 90+,$$

and the weight required for 1000 gallons is

$$\frac{9 \times 1000}{7000} = 12.9 \text{ pounds per day.}$$

It is advisable to have the soda introduced at regular intervals, say once an hour, during the day. For this purpose, the arrangement shown by Fig. 23 is convenient. The soda crystals are weighed out and dissolved in the bucket marked "receptacle for solvent in solution," and then when the pump is feeding water to the boiler, the stop valve under the bucket is opened cautiously, whereupon the solution will be drawn into the pump, and delivered to the boiler. Care must be taken not to admit air to the pump and disarrange its action. This same arrangement may be used to feed any desired solution into a boiler.

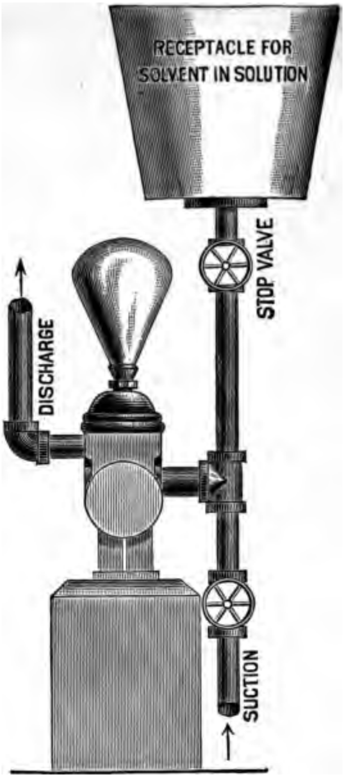

FIG. 23.

Soda ash is sometimes used instead of soda crystals, as it is cheaper and somewhat more efficient, on account of the caustic soda it may contain. Its chemical composition is uncertain, and it is therefore impossible to make satisfactory calculations for the quantity to be used. This, however, is commonly no real objection, for we seldom have a chemical analysis of the water, and cannot determine directly how much soda is required.

An excess of soda in a boiler is liable to cause foaming, and at high temperatures, corresponding to pressures now habitual for steam boilers, the soda is apt to attack the inside of water glasses; any indication of either action should raise the question whether too much soda is used, but the absence of such an indication does not show that we are using the right quantity. When a hard scale is formed by a water known to contain lime, we may infer that sulphates are present, and may find by trial the amount of soda to be used. Unfortunately, other impurities, such as organic matter, cause the formation of hard scale, and make this method uncertain. Such impurities often produce discoloration, and thus betray their presence. The deposits of lime, whether carbonates or sulphates, are commonly white or grayish, or sometimes fawn color.

It is sometimes proposed to use ammonium chloride, or sal-ammoniac, to break up lime compounds; in the first place, only the carbonates are acted upon by this reagent, and in the second place, the reagent itself, or the resultant chlorides, are liable to be broken up, giving free chlorine, which attacks the boiler.

Tannic acid, either commercial acid or in the crude state, may be used to break up a scale already formed; but as tannic acid does not decompose the sulphates, and as the compound of the acid with lime is not soluble, its use appears to be restricted. Many proprietary boiler compounds depend on tannic acid for their action. Acetic acid may also be used to break up the carbonates, but it likewise has no action on the sulphates; the carbonates are changed into soluble acetates, and can be blown out. Both tannic acid and acetic acid attack iron, but are not so dangerous as sulphuric or hydrochloric acids, which are sometimes recommended for breaking up scale. When a scale is once formed the safer way is to remove it with proper chipping and scaling tools; but this will be found to be impossible for many types of boilers unless they are largely dismembered for that purpose.

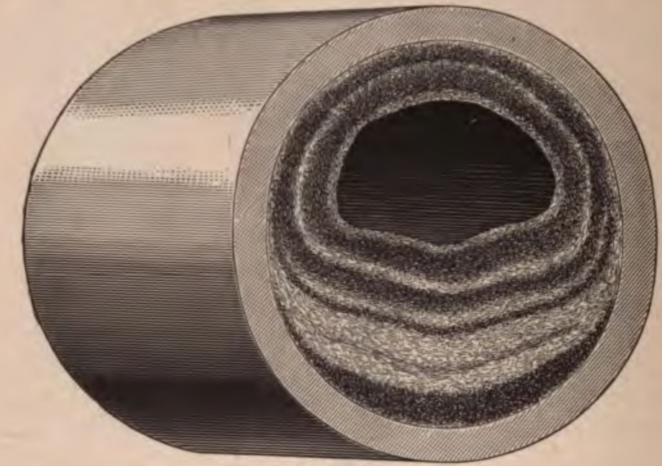

FIG. 24.

When river water is used in boilers, various earthy impurities are liable to be carried into boilers, such as clay and sand, together with soluble matters. Even waters from ponds or wells may contain considerable matter in suspension. Such substances can sometimes be removed by filtering or by allowing the water to stand so that the insoluble matter may be deposited. Very commonly a systematic blowing out from the surface of the water and the bottom of the boiler will remove such impurities from the boiler. If, however, lime and magnesium carbonates and sulphates are present, suspended matter is carried into the scale, and the scale may be made more troublesome in consequence. The carbonates are more likely to form a hard scale if any binding material, such as clay, is present.

Figure 24 shows the section of a feed pipe which was nearly choked with scale from lime water. Though the deposit of scale in a horizontal piece of feed pipe where the water may be heated by conduction and otherwise, especially during intervals of feeding, is probably more rapid than in the boiler itself, this may serve to call attention to the extent to which scaling may occur when precautions are not taken.

Lime Extracting Feed-Water Heater.—If water containing lime and magnesia salts is heated to a sufficient temperature before feeding it to a boiler, the deposits may be caught in the apparatus for that purpose. Figure 25 represents the Stillwell feed-water heater and purifier. The water is brought in through the pipe *X*, and flows over a series of corrugated plates *B*. Steam from the exhaust pipe of a non-condensing engine enters the heater at *L*, passes among the plates, and passes out at an orifice not shown in the cut. The calcium carbonate and suspended impurities collect on the corrugated plates, and the water flows down behind the plate *C* to the space *D*, and thence up through a filter chamber filled with hay or similar material. At *F* is a cup into which the feed water flows from the filter, and from whence it is drawn by the feed pump. The chamber *D* has a pipe to draw off sediment as it collects, and at *H* there is a drip from the steam chamber. A float in the float chamber *M* regulates the supply of entering feed water so as to keep a constant level in the heater.

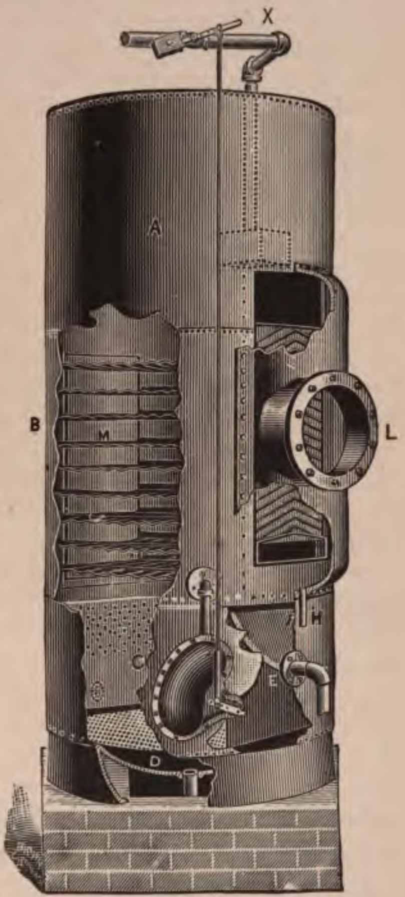

FIG. 25.

Calcium sulphate will not be deposited with any certainty in the open heater shown by Fig. 25. When the water contains the sulphate a closed heater, similar to this in principle, is used, but it is supplied with steam at full boiler pressure, and thus a temperature is easily reached at which all the lime salts are thrown down. Such steam, moreover, does not contain oil from the engine.

Sea Water. — The following table gives an analysis of sea water by Professor Lewes of the Royal Naval College, together with an analysis by him of a typical boiler deposit from a marine boiler: —

SALTS IN SEA WATER AND COMPOSITION OF MARINE BOILER SCALE.[1]

	Sea Water. Grains per Imperial Gallon.	Marine Boiler Scale. Per Cent.
Calcic carbonate (chalk)	3.9	0.97
Calcic sulphate (gypsum)	93.1	85.53
Magnesic sulphate	124.8	. . .
Magnesic chloride	220.5	. . .
Magnesic hydrate	. . . .	3.39
Sodic chloride (salt)	1850.1	2.79
Silicia (sandy matter)	8.4	1.1
Moisture	. . .	5.9

The three principal constituents of the marine scale are calcic sulphate, calcic carbonate, and magnesic hydrate, of which the first forms the greater part of the scale.

The calcic carbonate is kept in solution by the carbonic acid in the sea water, just as in the case for fresh water containing carbonate of lime, and is deposited when the carbonic acid is driven off by heat. There is, however, a reaction between the calcic carbonate and magnesic chloride at the temperature and pressure in the boiler, giving a deposit of magnesic hydrate and leaving calcic chloride in solution; so that only part of the calcic carbonate appears in the scale, and on the other hand, we may thus account for the presence of the magnesic hydrate in the scale.

The calcic sulphate forms so large a part of the scale, that we will give attention to it only in the further discussion. Calcium sulphate is more soluble in water at 95° F. than at any temperature higher or lower; and the solubility decreases with the rise of temperature, till at about 280° F., which corresponds to 50 pounds pressure absolute to the square inch, or 35 pounds above the atmosphere, the entire amount of calcium sulphate is deposited. In the early history of the marine engine, when low pressures of steam prevailed, we find jet condensers in use, and the boilers, which were fed from the brine in the hot well, were kept fairly free from scale by blowing out the concentrated brine. It was then customary to supply half again as much feed water as was evaporated, the excess being compensated by the concentrated brine blown out, and the water in the boiler had three times the degree of concentration found in the sea. As high pressure steam came into use, surface condensers became indispensable. When surface condensers first came into use, the waste of steam from leakage and otherwise was made up from water taken from the sea, with the result that the boilers gradually accumulated a heavy dense scale. Since it is customary to have an auxiliary boiler, called a donkey boiler, on steamships, the first device to avoid the scaling from the use of sea water in the main boilers appears to have been to supply the loss of steam from the donkey boiler, which was fed from the sea. This of course only transferred the difficulty from one place to another, even though a less objectionable one. At present the loss is made up by vaporizing sea water in a special boiler, which is heated by steam coils supplied with steam from the main boilers. The pressure may be low enough in this vaporizer to avoid the total precipitation of the calcium sulphate, and the brine may be kept at any desirable degree of saturation by blowing out, as in the early marine practice; and further, the vaporizer is so made that the steam coils may be readily cleared from scale.

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[1] Trans. Inst. Naval Arch., Vol. xxx., p. 330.

It should be pointed out that the decomposition of the calcium sulphate in sea water by the aid of soda is impracticable, on account of the large quantity of magnesium carbonate thrown down from the reaction on the magnesium sulphate.

A boiler fed with water condensed in a surface condenser, as is now common in marine practice, is liable to two difficulties: (1) the distilled water is apt to corrode or pit the plates of the boiler, and (2) the cylinder oil used in the engine is liable to be carried over into the engine and form oily scales and deposits.

When sea water is used in the boiler, either as the main boiler feed or merely to supply the waste, the boiler plates are protected by the scale of calcium sulphate, and general corrosion or local pitting is seldom troublesome. When care is taken to avoid the use of salt water, supplying the waste with fresh water from a distiller or otherwise, general corrosion and local pitting have both been found to occur to a dangerous degree. A simple remedy would appear to be to form a very thin scale by the use of sea water, and then avoid further use of sea water. It is, however, found that water from a surface condenser will gradually dissolve off such a scale, and it must be occasionally renewed. The care and intelligence required for such a method of protection may be more than can be expected in ordinary practice. There is also an objection to the introduction of any lime compound into a boiler, as will appear in the discussion of the difficulty from the collection of oil in the boiler. In both the American and the English navies it is customary to use slabs of zinc to protect the boiler plates from corrosion. The zinc is fastened to, or hung from the boiler stays, with which metallic connection should be made to ensure galvanic action. The zinc is gradually consumed, and becomes soft and friable, so that the slabs require renewal. It is recommended to supply ¼ of a pound of zinc for each square foot of grate surface.

It is a familiar fact that the cylinders of an engine may be oiled by introducing the oil into the supply pipe, and that the oil will be carried quite thoroughly over the surface of the cylinder by the steam; and, further, that the oil is carried out of the cylinder by the steam, and will appear in the condensed water in the hot well. It is evident that any oil is liable to be injurious if it gets into a boiler. It is, consequently, customary to filter the water from a surface condenser, to remove the oil as far as possible. For this purpose sponges have been used in the navy; they, of course, must be occasionally taken out and washed free from oil. A very simple and efficient filter has been made in the form of a rectangular box, with perforated plates near the ends; the water from the hot well runs into one end compartment, passes through a mass of hay in the middle compartment, and is drawn from the further end compartment by the feed pump. When the hay becomes foul it is thrown away, and fresh hay is put in. Professor Lewes advises for a filter a long tube filled with charcoal about the size of a walnut; of course the charcoal should be renewed when necessary. It cannot be expected that any system of filtering will remove all the oil from the water, but the larger part may be removed. It is advisable that no more oil than necessary shall be used in the cylinders of the engine, but the exclusion of oil from the cylinders, though recommended for some torpedo-boat engines, is not practicable in general practice.

Professor Lewes[1] gives the following account of an investigation of the collapse of the furnace flues of a large Atlantic steamer, which made the voyage in twelve days.

The boilers were five and a half years old, and were refilled with fresh water at the end of each voyage, while the waste of the voyage was made up by the use of about 70 tons of fresh water, but during the last voyage sea water was used for this purpose. Every four hours, while under steam, four pounds of soda crystals were put in the hot well, making two hundredweight during the run, the total capacity of the boilers being 81 tons. For lubricating purposes seven pints of valvoline were used in the cylinders every four hours.

When in port the boilers were allowed to cool down, and the water was run off and they were swept down with stiff brushes, and were afterwards sluiced out with a hose shortly before being filled with fresh water. No trouble occurred until five voyages before the final collapse, when some of the furnaces began to creep in; they were stiffened with rings and stays; and on succeeding voyages the whole of the furnaces got out of shape one after the other. Examination showed that they had never been very heavily scaled. On the furnace crown there was only a slight white scale not more than $\frac{1}{64}$ of an inch thick, while on the bottom of the furnaces there was a brown oily deposit $\frac{1}{16}$ of an inch thick, which in other parts of the boiler increased to $\frac{1}{8}$ or $\frac{3}{16}$ of an inch.

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[1] Trans. Inst. Nav. Arch., xxxii., page 67.

The valvoline was a pure mineral oil with a specific gravity of 0.889 and a boiling point of 371° C.

The composition of scales from several parts of the boiler is shown in the following table: —

COMPOSITION OF DEPOSITS IN A MARINE BOILER.

	From Top of Furnace.	From Bottom of Furnace.	Scale on Tubes.	Deposit above Scale on Tubes.	Deposit from Bottom of Boiler.
Calcic sulphate	84.87	59.11	50.92	11.60	22.52
Calcic carbonate	5.90	6.07	4.18	0.82	. . .
Magnesic hydrate	2.83	11.29	14.12	22.21	7.09
Iron, alumina, silica	2.37	2.85	7.47	9.14	34.85
Organic matter and oil . . .	3.23	19.54	21.06	50.20	27.95
Moisture	0.80	1.14	1.17	4.23	5.79
Alkalies	. . .	. . .	1.08	1.80	1.80

Careful examination of the organic matter and oil in these deposits showed that half of it was valvoline in an unchanged condition, which had collected around small particles of calcic sulphate.

All the deposits were rich in oily matter except the top of the furnaces, *i.e.* the place where the collapse occurred. There the scale was not only nearly free from oil, but perfectly harmless both in quantity and quality. It appeared entirely improbable that the scale on the top of the furnaces could be in its original condition.

When oil has entered a boiler the minute globules, if in large quantity, coalesce to form an oily scum on the surface, or if in small quantity remain in separate drops, but show no tendency to sink on account of their low specific gravity. They, however, come in contact with solid particles of calcic sulphate, coat them with oil, and so the light oil becomes loaded till it is easily carried along by convection currents and adheres to surfaces with which it comes in contact, which may be quite as likely to be the under surfaces of tubes as the upper surfaces. Since some brine is liable to find its way to the boiler, from leakage into the condenser or otherwise, even when sea water is not used directly, this action will occur in a boiler supposed to contain fresh water only.

The deposits thus formed are very poor conductors of heat, and the oily surface interferes with contact with water. On the crown of the furnace this soon leads to overheating of the plates, and the deposit begins to decompose, the lower layer in contact with the plate giving off gases which blow up the greasy layer, ordinarily only $\frac{1}{64}$ of an inch thick, to a spongy mass $\frac{1}{8}$ of an inch thick, which, because of its porosity, is even a better non-conductor of heat than before, and the plate becomes heated to redness and collapses. During the last stages of this overheating, the temperature has risen to such a point that the organic matter, oil, etc., in the deposit burns away, or is distilled off, leaving behind, as an apparently harmless deposit, the solid particles round which it had originally formed.

Such a deposit is more likely to be produced in boilers containing fresh or distilled water, as the low density of the liquid enables the oily matter to settle more quickly, while with a strongly saline solution it is very doubtful if this sinking point would ever be reached; it is evident also that when oil has found its way into the boiler, and is causing a greasy scum on the surface, the most fatal thing that can be done is to blow off the boilers without first using the scum cocks, because as the water sinks, the scum clings to the tops of the furnaces and other surfaces with which it comes in contact, and, on again filling up with fresh water, it still remains there, causing rapid collapse. A very remarkable instance of this is to be found in the case of a large vessel in the eastern trade, in the boiler of which an oil scum had formed. The ship having to stop some days in Gibraltar, the engineer took the opportunity of blowing out his boiler and refilling with fresh water, with the result that before he had been ten hours under steam the whole of the furnaces had come in. Under some conditions the oil-coated particles coalesce and form a sort of floating pancake, which, sinking, forms a patch on the crown of the furnace at one particular spot, and under these conditions the general result is the formation of a pocket.

A curious fact is that these oily deposits are found to contain a considerable amount of copper. Even mineral oils have a solvent action on copper and its alloys, and it is evident that the copper in the oily deposits has been obtained from the fittings of the cylinder and condenser. Fortunately, this copper is protected by oil, otherwise serious galvanic mischief would result.

Professor Lewes found from experiment that a coating, $\frac{1}{16}$ of an inch thick, of the oily deposit found in the bottom of a boiler, applied to the inside of a clean iron vessel, very greatly retarded the transmission of heat from a Bunsen flame, as shown by the time required to heat a known quantity of water to boiling point. Using an atmospheric blowpipe, he succeeded in raising the outside surface of the vessel, when coated with $\frac{1}{16}$ of an inch of the deposit, to the temperature of the melting point of zinc, and with an oxy-coal-gas flame he fused a hole in the bottom of a thin wrought-iron vessel thus coated and filled with water.

He further says that cylinders should be sparingly lubricated with a pure mineral oil having a high boiling point, and that animal or vegetable oil should never be used, because they are decomposed by the action of high pressure steam, producing fatty acids that attack iron, copper, and copper alloys.

Professor Lewes has proposed that marine boilers at sea shall have the water supplied with brine from which the lime compounds have been precipitated in a closed receptacle by the combined action of heat and soda crystals. The resulting brine contains mainly sodium and magnesium chlorides and magnesium sulphate, which do not form scale even though the concentration is carried to a higher degree than would occur from the supply of the waste of the boiler in this way for a voyage of some length. This method has not as yet been adopted in practice. Attention is called to the fact that an excess of soda should be avoided, since it would cause a bulky deposit from the action on the magnesium sulphate brought in by leakage of sea water into the condenser. A description of the apparatus for producing this brine without lime salts is given in the "Transactions of the Institution of Naval Architects" (see the reference, page 40).

Organic Impurities. — Water for feeding boilers, unless taken from a contaminated source, seldom contains much organic matter. Surface water from rivers or ponds may contain some vegetable matter, but if there are no other impurities such organic matter will not cause much trouble unless it is allowed to accumulate. The vegetable and other organic impurities commonly float on the surface of the water, when the boiler is making steam, or are carried around by convection currents, and may be blown out through a surface blow-out, like that shown by Fig. 26. It consists of a flattened bell or cone of sheet metal extending across the boiler at the water level, and turned so that the convection currents will carry and lodge floating substances into the mouth of the bell. The valve in the pipe leading from this bell may be opened from time to time to blow out the substances collected in it.

When a boiler has been at rest for some time, over night for example, the various solids in the boiler, if heavy enough, will settle to the bottom, and may be advantageously blown out before starting the boiler into action again. This may be accomplished by opening the blow-out valve or cock for a short time, until the water level falls a few inches.

Water from bogs frequently contains vegetable acids that are likely to corrode the plates of the boiler; in such cases soda crystals may be

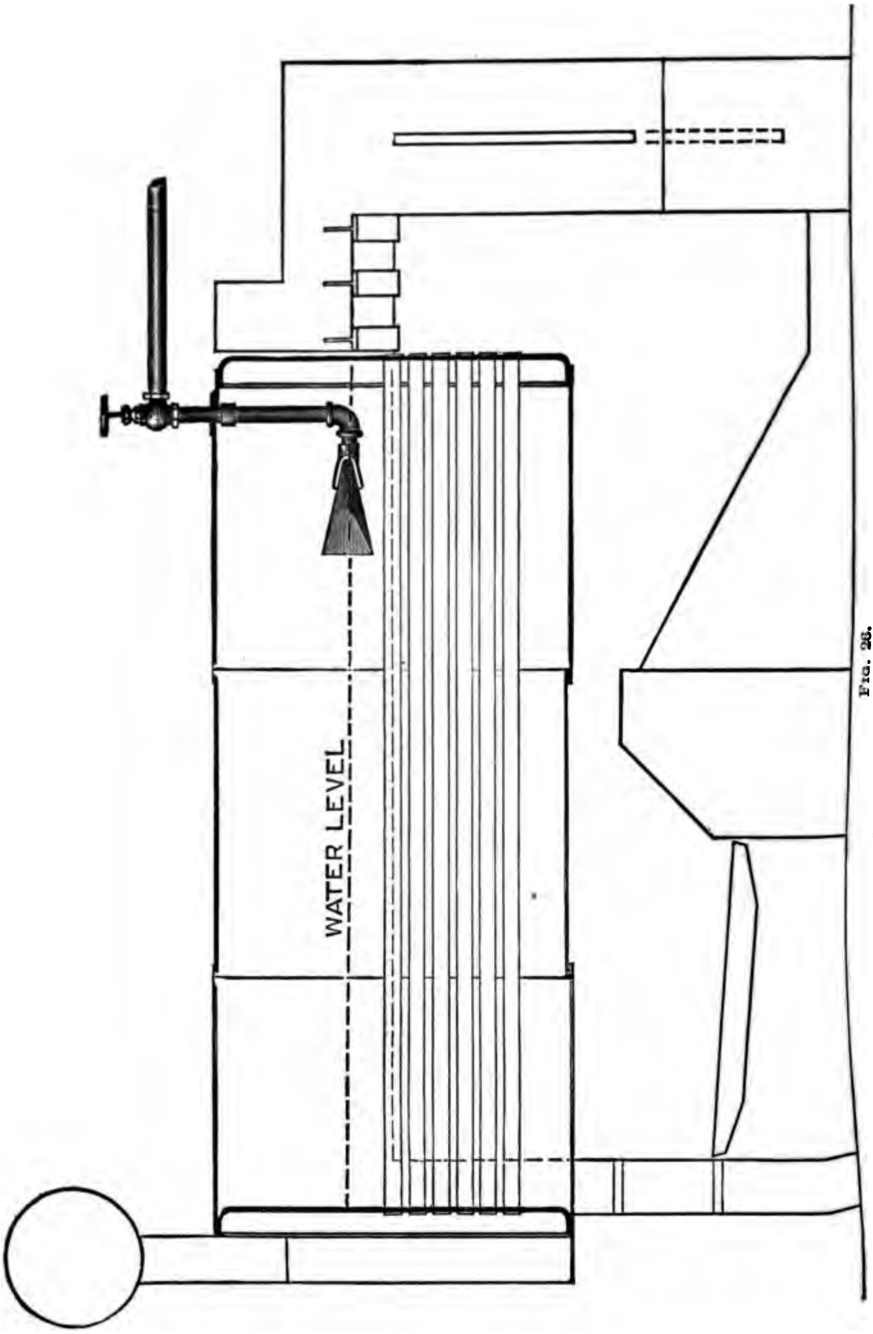


FIG. 26.

used to neutralize the acids; the proper amount must be found by trial.

The oil used in the engine is liable to get into the boiler if surface condensing is made use of; this subject has already received attention in connection with the discussion of marine boiler incrustations. Surface condensers are not commonly used in land practice, but very commonly the exhaust steam from non-condensing engines is used for heating in radiating coils, and there is an apparent gain from the use of the warm water from the return pipes. This water is, however, liable to be contaminated by oil, and the oil when it gets into the boiler may cause serious damage, such as was found to occur in marine boilers. If the feed water has a little vegetable matter in it, the effect of the oil is much worse than if the water supply is pure. Again, the oil is very troublesome if the water contains lime salts. The bad effect of oil or other impurities on lime scale has been already noted. Usually it will be found better to reject the water returned from a heating system supplied with exhaust steam, as the apparent economy is liable to be more than counterbalanced by damage to the boiler. The externally fired tubular boilers commonly used in this part of the country are liable to bulge in the sheets over the furnace, as shown in Fig. 27, if oil gets into them. When the plate is cut out, a hard deposit of oil, commonly mixed with other impurities, will be found adhering to the plate; this deposit is a very poor conductor of heat, and it causes so much overheating of the plate that it bulges out under the pressure of the steam.

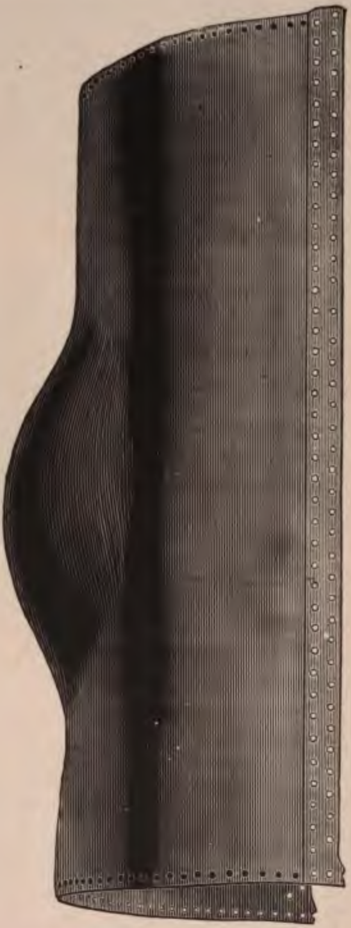


FIG. 27.

In isolated cases it will be found that water of a stream may be so contaminated with chemicals from some industrial establishment, that it acts energetically on the boiler plates; in such case the water must be abandoned unless the contamination can be stopped.

Kerosene and Petroleum Oils. — Both crude petroleum and refined kerosene have been used in steam boilers to mitigate the effect of incrustations of calcium carbonate and calcium sulphate. From what is known of the bad effects of the heavier petroleum products, such as the mineral oils used for lubricating steam-engine cylinders, it appears to be unwise to introduce crude petroleum into a steam boiler. The same objection does not apply to refined kerosene which is not known to have any bad effect in a boiler. Both oils are said to change the deposits of lime from a hard scale to a friable material, which may be easily removed. It is further said that these oils will soften and loosen scale already formed. In one case 40 gallons of kerosene were used in 24 hours in the boilers of a steamer of about 3000 horse power. These boilers showed no incrustation, but considerable corrosion.

Corrosion is distinguished as general corrosion or wasting, pitting, and grooving.

General corrosion is difficult to detect, as it acts more or less uniformly over large surfaces, and even at riveted joints, the two plates and the rivet heads waste away equally, so that the thinning of the plates is not easily noticed. Old boilers not infrequently fail from general corrosion, and then are likely to fail in the plate rather than in the riveted joint where the double thickness of plate gives an advantage. Boilers that have been at work should have the plates below the water line drilled and the thickness measured; if the effective thickness of the plate is found to be much reduced, the working pressure should be made proportionately lower. Figure 28 shows an example of general corrosion, and Fig. 29 another, but complicated with cracking at the rivet holes. Both show the protection given to the plate by the rivet head, and one may readily see how the wasting of the rivet heads may be overlooked.

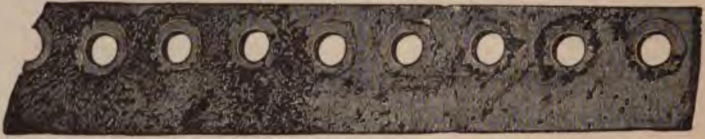


FIG. 28.

Pitting is likely to occur when the corrosion, either locally or over large surfaces, takes place rapidly. It appears to be due to lack of homogeneity of the metal of the plate, and sometimes appears to indicate galvanic action. Though every precaution to avoid galvanic action should be taken, it is better to assume damage to be due to such action only when there is direct evidence of its existence. Figure 30 shows pitting over a large surface, and Fig. 31 shows local pitting in the corner of a flanged plate with general corrosion of the flat surface of the plate. It is fair to assume that the disturbance

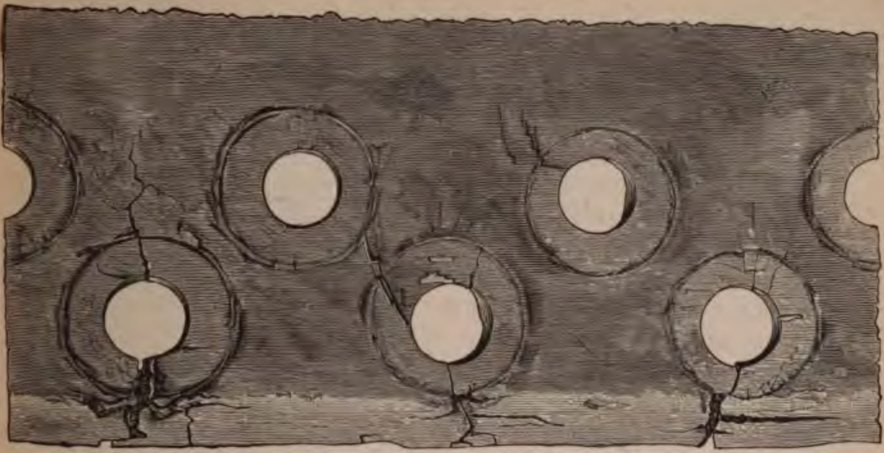


FIG. 29.

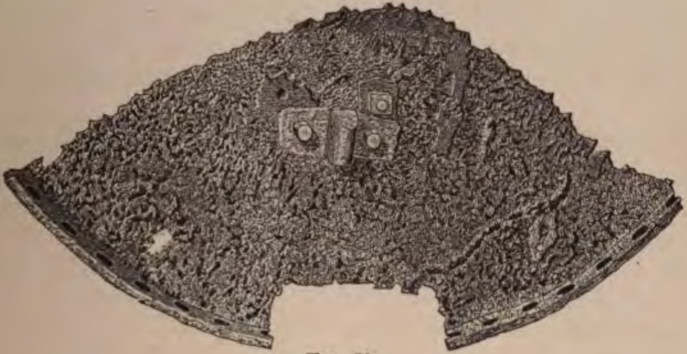


FIG. 30.

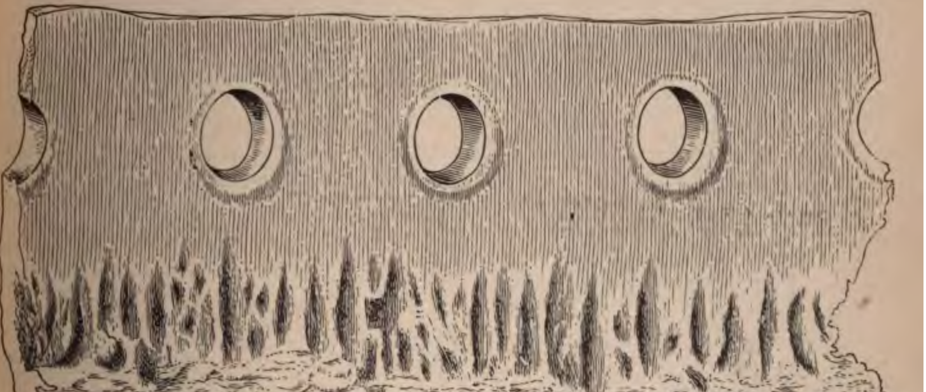


FIG. 31.

of the metal in the process of flanging may determine the vertical forms of the pitting. The horizontal plate shows irregular pitting.

Grooving is usually due to the combination of springing or buckling of a plate and local corrosion. The buckling may be due to insufficient staying; then the plate springs back and forth as the steam pressure varies. Or buckling may be due to improper staying or fastenings, which localizes the change of shape due to expansion. In either case the metal is fretted at the place where the greatest bending takes place, and very much weakened. A crack is liable to be formed which may grow wider and deeper till the plate shows signs of failure. Such cracks may be very narrow and difficult to find, but usually the fretting of the metal, whether a crack is formed or not, is accompanied by local corrosion, which makes a groove of some width. If the water used forms a scale on the boiler plates, the working of the metal throws off the scale and exposes the surface to the water so that corrosion takes place there, though elsewhere the plate is protected.

As one example of insufficient staying, we may take the flattened surface in a wagon-top locomotive boiler (Plate V.), where the barrel is expanded to join the shell over the fire box. The surface cannot be stayed from side to side for lack of space between the tubes, and is merely stiffened by riveting three pieces of T iron to the shell. In this case the T irons have through stays at their upper ends over the tubes. Grooving is liable to occur in this locality even when the plates are stiffened as shown.

Grooving from too great rigidity is liable to occur in the end plates of Cornish and Lancashire boilers (see pages 5 and 6). The long furnace flues expand more than the external shell, and expand more at the top than at the bottom, due to the heat of the furnace and of the gases in the flue beyond the furnace; and further, the circulation of water under the flues is likely to be imperfect, so that the bottom of the shell is not so hot as the top. These unequal expansions must be accommodated by the springing of the end plates, and if the springing is too much localized grooving is sure to occur. The furnace flues should be at least nine inches from the shell, and the end plates should be flanged where they are joined to the flues and shell, instead of using angle irons. The use of gusset plates for staying the ends of these boilers is likely to give too much rigidity and to localize the springing of the plates, unless care is taken to avoid it.

Grooving from either too great or too little rigidity can be avoided only by a proper design, which must be guided by experience. If a boiler shows defects of staying, it may be possible to put in additional stays after the boiler is completed and at work; or in some cases too great rigidity may be remedied by rearranging the staying. Such remodelling of a boiler is usually difficult and unsatisfactory.

Loss from blowing out Brine. — In the discussion of the use of sea water in boilers it was said that the old custom was to feed one and a half as much sea water from the hot well of the jet condenser as was evaporated, thereby raising the concentration of the brine to three times that of the sea, namely to $\frac{3}{32}$.

In calculating the loss of heat from blowing out hot brine, we may assume the specific heat of both the sea water and the hot brine to be the same as that of fresh water. Let the absolute pressure in the boiler be p, and let r and q be the heat of vaporization and the heat of the liquid corresponding; let the temperature of the water in the hot well be t_1, and let q_1 be the heat of the liquid. The heat required to raise one pound of feed water from the temperature of the hot well to the temperature in the boiler, and to vaporize it under the pressure in the boiler, will be

$$r + q - q_1;$$

this is the heat usefully expended. If n is the hot brine blown out for each pound of water vaporized (usually $\frac{1}{2}$ pound), then the heat required to raise the water blown out from the temperature in the hot well to that in the boiler is

$$n(q - q_1);$$

this is the heat lost by blowing out the hot brine. Of the heat applied to the boiler, we have the loss in per cent,

$$100 \frac{n(q - q_1)}{r + q - q_1 + n(q - q_1)} = 100 \frac{n(q - q_1)}{r + (n + 1)(q - q_1)}.$$

A considerable portion of the heat lost in the hot brine may be transferred to the feed water drawn from the hot well by the aid of a feed-water heater, and thus saved. A simple form of heater may be made by carrying the hot brine through a small pipe inside the feed pipe; the currents of water will naturally flow in opposite directions, and thus give the most efficient interchange of heat. If the hot well is near the boiler, the feed pipe may not be long enough to allow of this form of heater.

The density of brine in the boiler is ascertained by a salimeter, which is a form of hydrometer graduated to read zero in fresh water, $\frac{1}{32}$ in sea water, and the graduation is extended to give the density of brine in thirty-seconds, so far as may be needed. When jet condensers were used at sea it was customary to carry the density to $\frac{3}{32}$ only. With surface condensers the density is frequently carried as high as $\frac{6}{32}$; no inconvenience is found in this custom, and as less water is taken from the sea the formation of incrustation is less rapid.

Wrought-iron plates should show a limit of elasticity of 23,000 pounds, and a tensile strength of 45,000 pounds to the square inch.

Cast Iron in different forms will show a tensile strength of 12,000 to 20,000 pounds to the square inch. Gun iron, which is cast iron made with special care and skill from selected stock, has shown a tensile strength of nearly 30,000 pounds to the square inch. In compression the strength of small pieces may be as high as 80,000 pounds to the square inch, but longer pieces, like columns, fail at 30,000 pounds to the square inch.

Cast iron is used for some or all of the parts of sectional boilers, and for fittings such as manholes, though wrought iron is preferable for such purposes. Flat plates at the ends of cylindrical boilers are sometimes made of cast iron; for example, the flat ends of the cylindrical shells of the Babcock and Wilcox boilers (see Fig. 14, page 12) are made of cast iron. In general, cast iron should never be used when it is subject to severe changes of temperature or to stresses from unequal expansion, and should be replaced by wrought iron or mild steel whenever it is practicable.

Couplings, elbows, and other pipe fittings are made of cast iron. The brittleness is a convenience when changes are to be made, as joints that cannot be opened are readily broken.

Malleable Iron, which is cast iron toughened by being deprived of part of the carbon, is used for pipe fitting and for fittings of steam boilers. It is used in place of cast iron for sectional boilers and for parts of water-tube boilers. Though tougher than cast iron, and though it will endure forging to some extent, its variability in quality and unreliability prevent much reduction in weight and size when substituted for cast iron.

Copper is largely used in Europe for making fire boxes of locomotive boilers and torpedo-boat boilers. Its greater cost is in part offset by the value of the scrap copper after the fire box is worn out.

Copper for fire boxes, rivets, and stays should have a tensile strength of 34,000 pounds to the square inch, and should show an elongation of 20 to 25 per cent in 8 inches. It should not contain more than one-half per cent of impurities. The greater ductility of copper, and its greater thermal conductivity, permitting of greater thickness for furnace plates, recommends it to European engineers.

Copper washers, gaskets, etc., are used for making steam-tight joints.

Bronze and Composition. — Brass. — Bronze is properly an alloy of copper and tin; thus gun metal is 90 parts of copper to 10 of tin. Compositions of various qualities are made of copper and zinc with more or less tin. Brass is an alloy of copper and zinc; for example, brass smoke tubes are made of 70 parts of copper to 30 parts of zinc. Lead is added to brass and composition to reduce the cost and to make the metal work easier. It may be considered as an adulteration, as it cheapens the metal at the expense of the quality. There are many special bronzes, such as phosphor bronze and aluminium bronze, which are used for special purposes.

Brass is used to some extent for smoke tubes of locomotive and other boilers, on account of its greater thermal conductivity, by European engineers. In America, brass is used for valves, gauges, and other boiler fittings. Composition or bronze is advantageously used for the valves and seats of safety valves and wherever the service endured is exceptionally hard. Brass is more commonly used because it is cheaper. In a general way it may be said that the cost and quality of brass and composition is proportional to the copper it contains; thus red brass is better and costs more than yellow brass. Many small brass fittings on the market are sold at a price which precludes the use of proper alloys, and they are consequently soft and worthless.

CHAPTER V.

BOILER SETTING AND FURNACES.

THE boiler setting for a stationary boiler consists of the foundation and so much of the flues and furnace as are external to the boiler proper. The entire furnace of externally fired boilers is in the setting, and in some cases, as with the plain cylindrical boiler, the flues are also formed by the setting. Some internally fired boilers — for example, the Lancashire boiler — have flues in the setting in addition to the boiler flues; others, like the upright boiler (Fig. 6, page 7), have only a foundation. Locomotive boilers rest on the frame of the locomotive; they can scarcely be considered to have any setting. Marine boilers are seated on plates that are built into the framing of the ship.

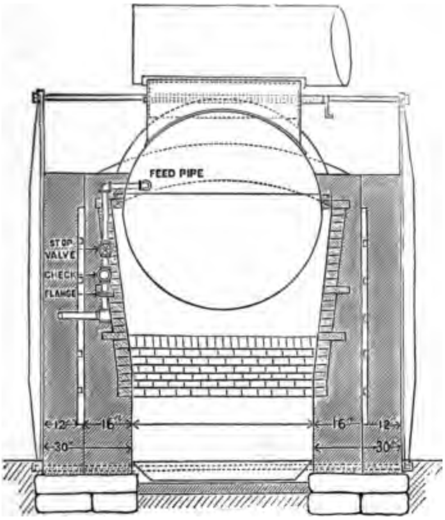


FIG. 32.

Cylindrical Tubular Boiler Setting. — This is clearly shown by Plates II. and III., and has already been referred to in the description of the boiler. As further illustration, we may refer to Figs. 32 and 33, showing a setting proposed by the Hartford Boiler Insurance Co. First we have the foundation of large stone blocks; then the inner walls, which in this case incline outwards, to give access of the hot gases to the sides of the shell. This wall is lined with fire brick in the furnace and a little beyond the bridge. The outer wall is distinct from the inner wall, to allow for unequal expansion. Bricks projecting from this wall at intervals afford a support to the inner wall. The space between the walls should be closed, since an air space is a good insulator only when there is no circulation through it. In this case, the side walls are carried only a little higher than the water line, and the top of the boiler must be covered with some non-conducting material, put on in blocks, or laid on in the form of a cement. The first method is preferable, as it allows examination of the shell of the boiler. On Plates II. and III. the side walls are carried to the full height of the boiler; with such a construction, radiation from the top of the boiler may be prevented by filling the space

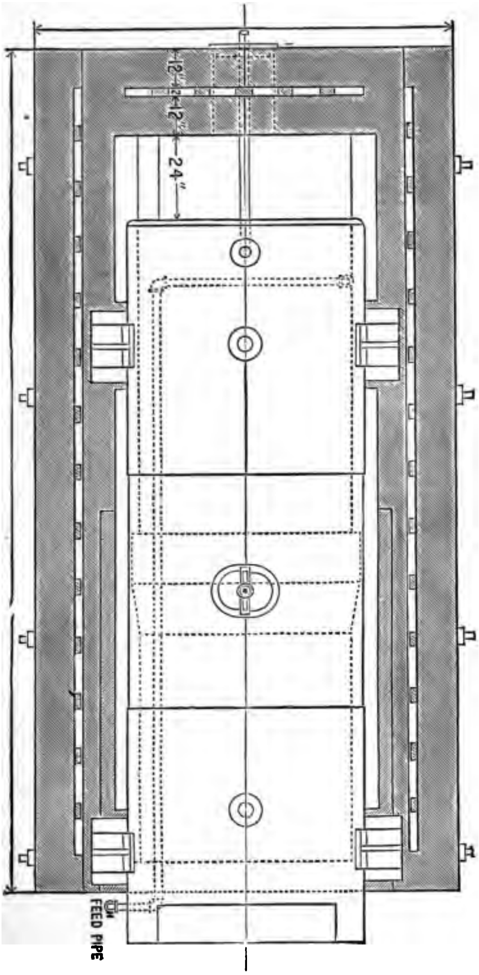


FIG. 33.

over the boiler with sand or ashes. Though this method is cheap and effective, and at the same time makes the clearing and inspection of the top plates of the boiler comparatively easy, it has the defect that the sand or ashes may harbor moisture when the boiler is not working and give rise to external corrosion. Coal ashes, being hygroscopic, are unfit for such purposes.

Figure 34 shows a setting with a brick arch over the top of the boiler, with an air space between the boiler and the arch.

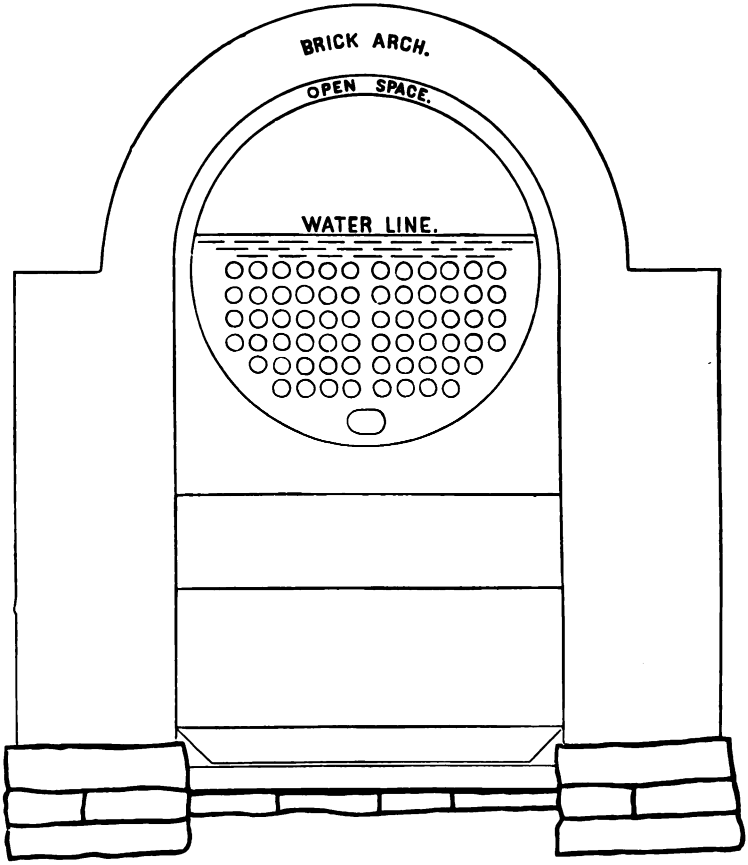


Fig. 34.

Figures 35 and 36 show the smoke box and uptake projecting in front of the cast-iron boiler front.

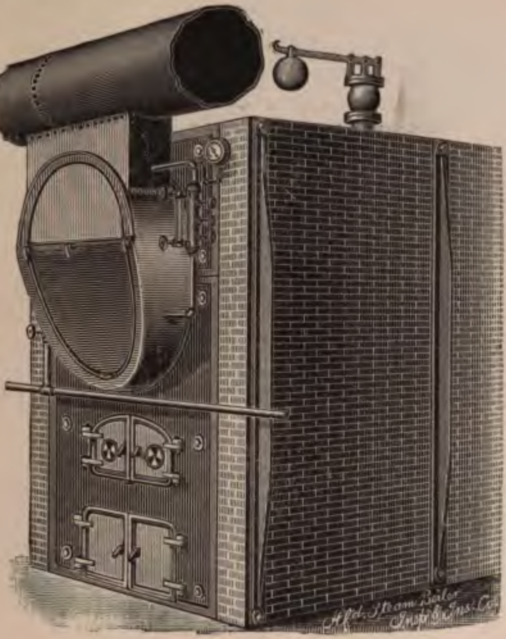


FIG. 35.

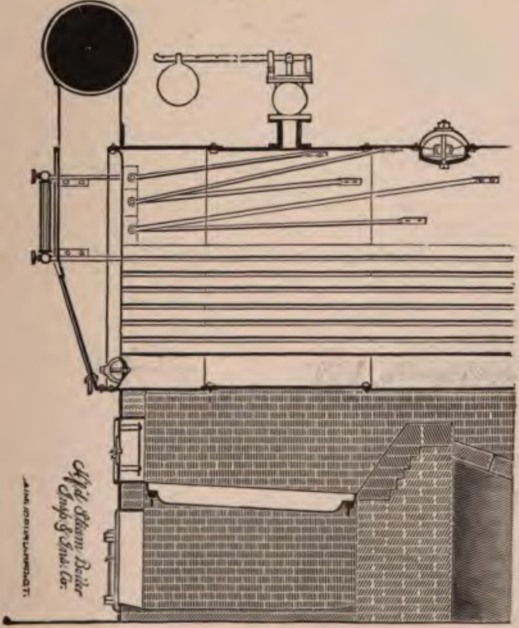


FIG. 36.

A flush boiler front with the smoke box and uptake is shown by Figs. 37 and 38.

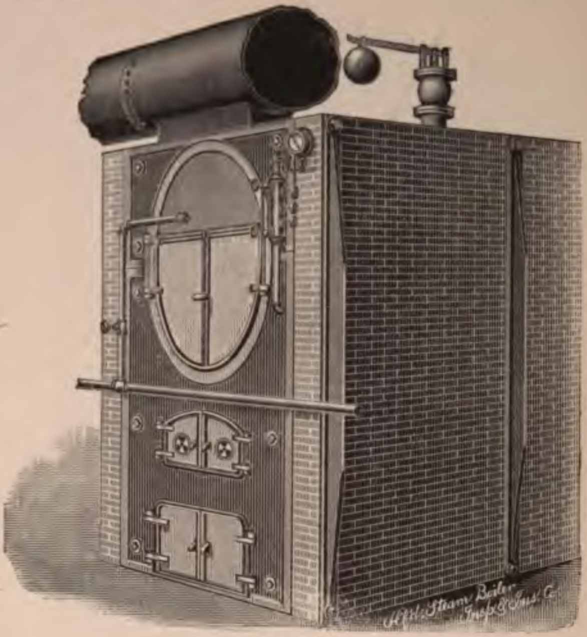

Hf'd Steam Boiler Insp & Ins. Co.

FIG. 37.

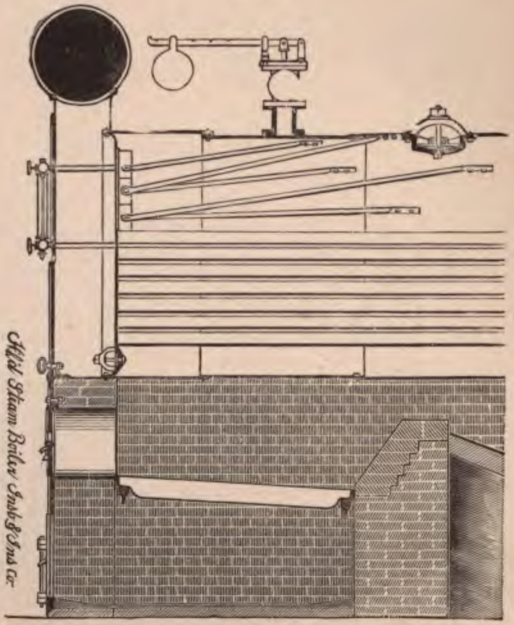


FIG. 38.

The setting for a two-flue boiler, or for a boiler with several large flues in place of the numerous fire tubes of the tubular boiler, is substantially the same as those just described.

Lancashire and Cornish Boiler Settings are arranged as shown by Fig. 4, page 6. The flues *a*, *b*, and *c* are lined with fire brick, and the boiler seat and tops of the side flues are made with blocks of fire clay· The draught is sometimes a *wheel draught*, passing forward from the ends of the furnace flues, through *c*, back through *b*, and again forward through *a*. Sometimes a *split draught* is used, passing forward through *c*, and then dividing it passes back through both *a* and *b*.

Settings for Water-tube Boilers, as shown by Fig. 14, page 12, and Fig. 16, page 14, resemble the setting for the cylindrical tubular boiler in external appearance. The furnace and bridge wall are also similar to those for the cylindrical boiler. Special bridges, extending among and across the tubes, are required to give the proper circulation of the products of combustion.

Water-tube boilers for marine work, like the Thornycroft boiler, shown by Fig. 20, page 15, and the Almy boiler, Fig. 21, page 16, are enclosed in a sheet-iron casing, lined with blocks of non-conducting material. Asbestos, or a compound of which magnesia is a principal ingredient, are commonly used. Fire brick and pumice stone are used with the Thornycroft boiler to intercept heat that would be radiated downward. The spaces in ships under boilers, being more or less inaccessible, and being subject to the influence of heat and moisture, are liable to show excessive corrosion.

Furnaces. — There are certain general conditions to which the construction of furnaces should conform if high efficiency is desired. Some of these depend on the requirements for good combustion, and some depend on the size, strength, and endurance of the human frame, since hand firing is almost universal. Some of these conditions are violated in the design and arrangement of furnaces in certain types of boilers; deviation from them involves either a demand for greater strength and skill on the part of the fireman, or a loss of efficiency of both.

These conditions, with examples of good and bad practice, are as follows: —

There should be an abundant and uniform supply of air to the under surface of the grate. About the only cases where this condition is not easily fulfilled is in the design of furnace flues of Lancashire boilers and Scotch marine boilers.

A small supply of air is required over the grate for burning smoky fuels like bituminous coal. This air is very commonly supplied through a circular grid or damper in the fire door. The fire door is commonly protected from direct radiation by a perforated wrought-iron plate, which also serves to distribute the air coming through this grid. Since the air thus supplied is cold it must be small in amount or it will chill the gases and check combustion instead of aiding it.

Leakage of cold air into the furnace, or into the combustion chamber or flues beyond the furnace, injures the draught and reduces the temperature of the products of combustion and is a direct source of loss. All externally fired boilers and water-tube boilers are liable to suffer from leakage of air. Locomotive and Scotch marine boilers are usually free from this defect.

The incandescent fuel on the grate should not come in contact with a cold surface. Furnaces lined with fire brick, such as are used for externally fired boilers, conform to this requirement. Vertical boilers, marine boilers, locomotive boilers, and all other boilers having the furnaces in fire boxes or flues violate this condition, as the plates in contact with the fire are kept nearly at the temperature of the water in contact with the other side, and are therefore much colder than the fire.

There should be an abundant opportunity for complete combustion of gases coming from the fuel with hot air drawn through the fuel, before the flame is chilled by contact with cold surfaces. This condition is best fulfilled by having a clear space over the grate. Externally fired boilers commonly have two feet or more between the grate and the boiler shell immediately over it, and combustion may continue beyond the bridge. Locomotive boilers have from four to six feet between the grate and the fire-box crown sheet, but the flame is quickly drawn into and extinguished by the tubes. To aid combustion and to protect the lower part of the tube sheet, a brick arch is frequently carried across the fire box over which the flame must pass on the way to the tube. The lack of space over the grate of flue furnaces, as in the Scotch marine boilers, is only partially compensated by the combustion chamber beyond the furnaces.

Loss from external radiation is almost entirely avoided in internally fired boilers. Externally fired boilers are subject to more or less loss from conduction and radiation.

The fire grate should not be longer nor wider than can be conveniently reached by the fireman in throwing on fuel and in cleaning the grate. A narrow grate should not be so long as a wide grate. In general, a hand-fired grate should not be more than six feet long, and if it is over four feet wide two fire doors should be provided. These conditions are usually fulfilled by the design of externally fired boilers, locomotive boilers, and water-tube boilers. Attention has been called already to the difficulty in getting proper space for the grates in flue furnaces. With the common diameters of the furnace flues, a length of five feet should not be exceeded. Flues in marine boilers have been made eight feet long; in such case the further end of the grate is sure to be inefficiently fired. To aid in firing and to use the space below and above the grate to the best advantage for the supply of air, and for combustion, the grate is commonly given an inclination downwards of about $\frac{3}{4}$ of an inch to the foot.

As an extreme example of deviation from these proportions, we may cite the Wooten locomotive fire box, designed to burn anthracite slack. The grate is made eight feet wide and twelve feet long.

For convenience in throwing on coal and in cleaning the grates, the floor on which the fireman stands should be about two feet below the grate. This can usually be arranged for stationary boilers. The grate of a locomotive is commonly below the floor of the cab; this facilitates throwing on the coal; some form of rocking grate is used to shake down the ashes. The side furnaces of Scotch marine boilers are commonly too high for convenient firing, and the middle furnaces may be too low for convenience in cleaning the grate.

Excessive heat in the fire room should be avoided as far as possible; the labor of feeding and cleaning a furnace for rapid combustion is always severe, and when combined with great heat, it rapidly exhausts the fireman. If land boilers are properly clothed to avoid radiation, and if the fire room is airy and well ventilated, the heat will not be excessive. It is, however, very difficult to avoid excessive heat in the stoke hole of a steamship. Of course the radiation from the glowing fuel when the fire doors are open cannot be avoided, but it ought to be possible to clothe the fronts of marine boilers more perfectly than is now the common practice. Moreover, the ventilation of the stoke hole is commonly defective; the air pours down through the ventilators and makes cold spots immediately beneath them, while other parts of the stoke hole are hot. Forced draught with closed stoke hole usually gives good ventilation; with closed ash pit it is liable to give defective ventilation.

Grate Bars are commonly made of cast iron, as it is cheaper and lasts as well as wrought iron. Sometimes wrought-iron bars are used on locomotives and elsewhere, if they are expected to withstand rough usage.

Cast-iron fire bars are generally $\frac{5}{8}$ to one inch thick at the top, and $\frac{5}{16}$ to $\frac{5}{8}$ of an inch thick at the bottom; they are about two inches deep at the ends, and three to five inches deep at the middle. To provide for wasting of the upper surface, they are made full width for some distance down from the top, thus forming a sort of head; then they are rapidly narrowed down to a web that is tapered gradually toward the bottom. The space between the bars depends on the draught and the nature of the fuel; with ordinary coal and natural draught, $\frac{3}{8}$ to $\frac{1}{2}$ of an inch is allowed. Lugs or projections are cast at the ends and at the middle, so that the bars shall be properly spaced when laid side by side. With forced draught, the bars may be $\frac{3}{8}$ to $\frac{9}{16}$ of an inch wide at the top, and the distance between the bars may be $\frac{1}{16}$ to $\frac{1}{4}$ of an inch. A dead plate two inches wide should be fitted to the furnace tube of marine boilers to prevent admission of air at that place.

The length of fire bars should not exceed three feet; the length of a fire grate may be made up of two or three short bars. Bars are commonly cast in pairs, or three or four may be cast together to resist twisting and warping under heat.

Wrought-iron fire bars are formed with a head and web, but are of uniform depth, as they are cut from a rolled bar; they are bolted together in sets of six, with washers to give the proper spacing. For marine boilers they may be $\frac{5}{16}$ of an inch thick at the top, with spaces $\frac{3}{16}$ of an inch wide, or less.

Rocking Grates have the grate bars pinned to short arms or rockers on transverse shafts at the ends; there are two sets of such rockers: alternate bars, or pairs of bars, being pinned to different sets of rockers, so that when the grates are rocked with a hand bar, the alternate bars (single or double) move up and down, back and forth, and thus break up the clinker and shake out the ashes. The bars, commonly, are of special shape, with interlocking side projections that assist in grinding up the clinker. They are used on stationary and locomotive boilers, but have not been adapted to marine boilers.

Firing. — Care, skill, and intelligence are required to burn coal rapidly and economically. There is a marked difference in the ability of trained firemen to make steam with a given boiler, and probably there is nothing more wasteful and costly than a poor or careless fireman.

The method to be adopted in firing depends on the type of boiler, the kind of coal, and the rate of combustion. Three methods of firing may be distinguished: —

Spreading, which consists in distributing small charges of coal evenly over the surface of the fire at short intervals. In this method the object is to deliver the coal just where it is wanted, and then not disturb it. The fire can then be kept in just the right condition at all times, and probably the best results can be thus obtained, both in absolute quantity of steam and in economy, provided the coal used is well adapted to this method. Care must be taken to have the door open as little as possible, or an undue amount of cold air will be admitted through the fire door.

Anthracite coal should always be fired by spreading, and should be disturbed as little as possible after it is thrown in place. Unless the fire is urged very little clinker will be formed, and the ashes are readily shaken out by a pick or hook run up between the fire bars. The thickness of the fire may vary from four to eight inches, depending on the size of the coal and the strength of the fire.

Dry bituminous coal, and other bituminous coals, if not very smoky and if in small pieces, can be advantageously fired in this way. Each shovelful thrown on will give off volatile matter which will burn with the excess of air coming through the fuel, and very little smoke will result.

Figure 39 shows the appearance of the furnace of an externally fired boiler, with the spreading method of firing.

Side firing consists in covering all of one side of the fire with fresh fuel, leaving the other bright. The smoke given off from the fresh fuel can then be burned with the hot air coming through the bright fire. This method of firing is best carried on with two furnaces leading to a common combustion chamber; the furnaces are fired alternately, at regular intervals, with moderate charges of coal. It is customary to admit air through the grid in the fire door when the fuel is giving off gas.

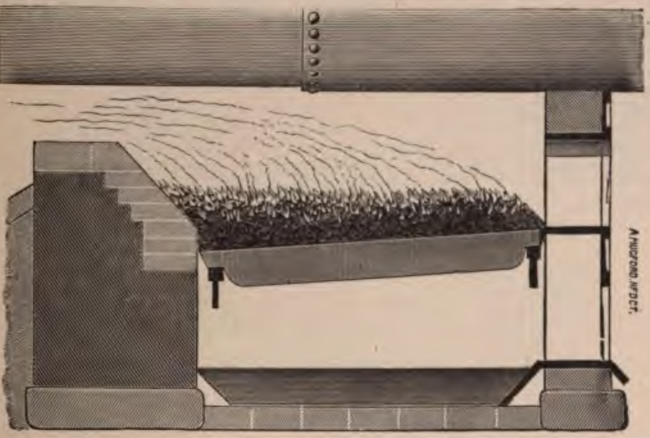


FIG. 39.

Coking the coal on a dead plate, or on the grate just inside the fire door, is perhaps the best way of burning a smoky coal. The appearance of the furnace is shown by Fig. 40.

The volatile products driven off from the heap of coal near the furnace door burn with the hot air coming through the clear fire at the rear. As soon as the charge is coked it is pushed back and spread over the grate, and a new charge is thrown on.

With bituminous coal the fire should be thicker than with anthracite coal; from 8 to 12 inches gives good results.

The method too often followed by ignorant and indolent firemen, of throwing on as much coal as the furnace will hold, and then sitting down to wait till the steam pressure falls, needs to be mentioned only to condemn it.

Mechanical Stokers, feeding coal regularly from a hopper, have been invented in a variety of forms from time to time. Since the hopper may be made of considerable size, manual handling of the coal may be entirely avoided, and one man can easily attend to a number of furnaces with little labor and exposure to heat. It would appear also that a more even and better regulated combustion may be had than with hand firing.

All such devices, which depend on having moving parts inside a confined furnace, quickly get out of order through the combined action of heat and dust.

The Roney stoker shown by Fig. 41 appears to have given good results when applied to externally fired boilers.

The grate bars extend across the furnace and form a series of steps hung on pivots near the top, as shown more clearly in Fig. 42 and Fig. 43. The lower edge of the web of the grate bar has a rounded lug at the end that rests in a corresponding groove across the rocker bar. The rocker bar has a slow and regular reciprocation communicated to it by the connecting rod which takes hold of the agitator, which in turn derives its motion through a link from the disk crank that is driven by a small steam engine. The coal in the hopper slides down onto the feed plate which is hung at the outer edge to the pusher, and the pusher is driven by the agitator; a feed wheel regulates the action of the pusher and feed plate, and thus regulates the rate of feeding the coal. The coal slides slowly over the dead plate onto the grate. The grate bars rock back and forth from the inclined position shown by Fig. 42 to that shown by Fig. 43, and the coal gradually works down the grate, burning as it goes. The combustion is supposed to be regulated so that nothing but ashes and clinker remain at the foot of the grate. The unburned refuse collects

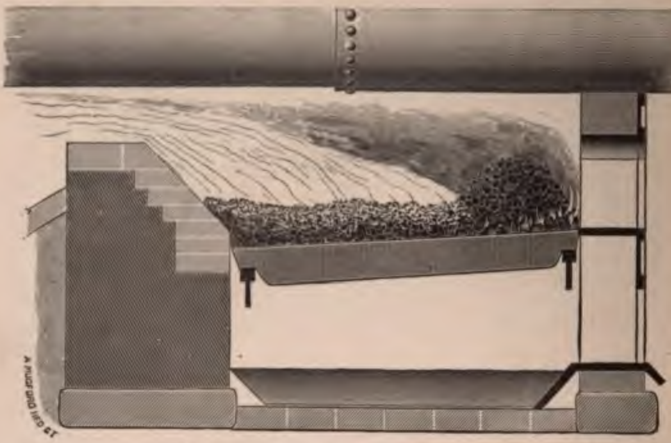
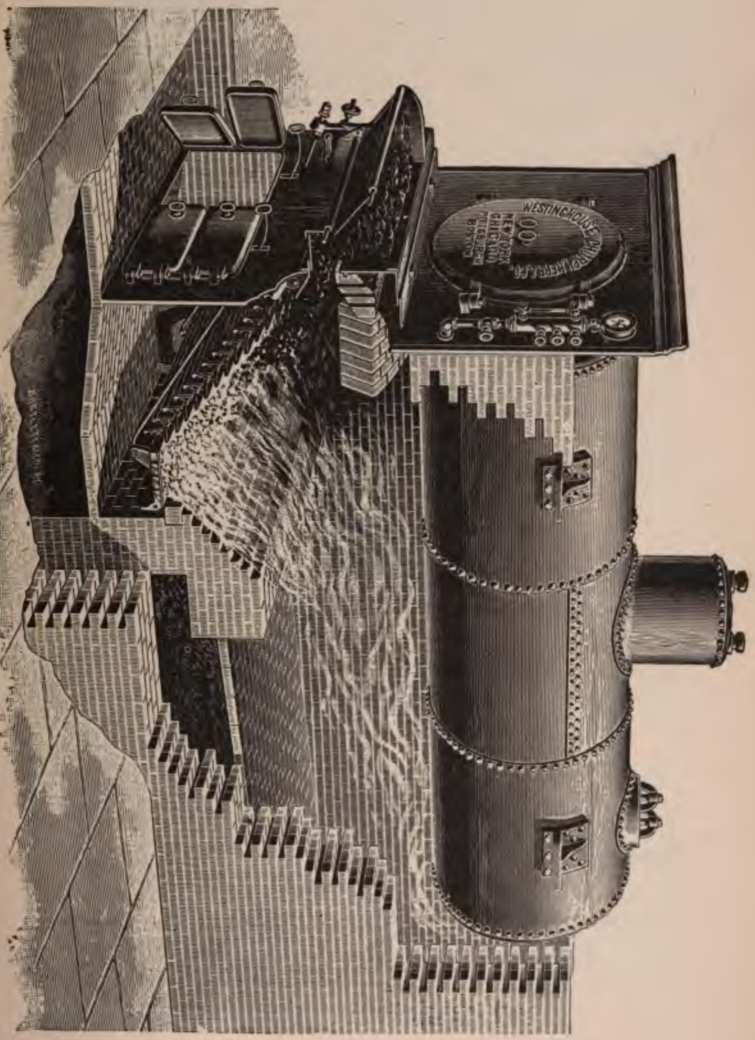


FIG. 40.


WESTINGHOUSE CHURCH KERR & CO.
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NEW YORK
CHICAGO
PITTSBURGH
BOSTON

Fig. 41.

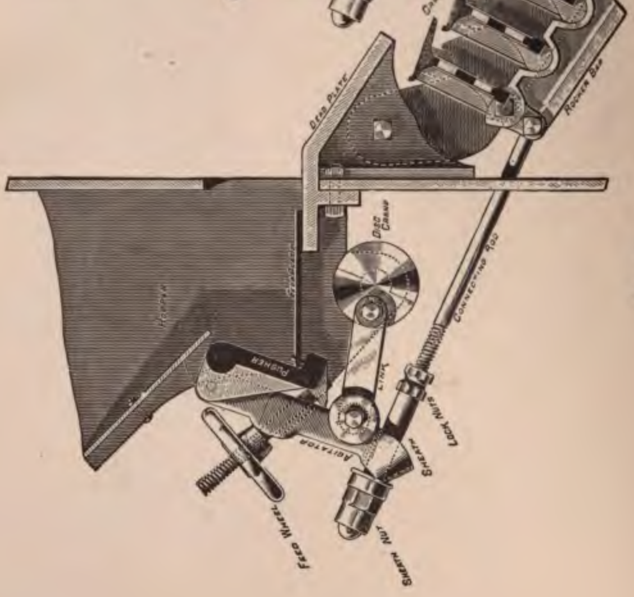
Feed Wheel
Sheath Nut
Agitator
Sheath
Lock Nuts
Pusher
Link
Hopper
Connecting Rod
Disc Crank
Dead Plate
Grate Bars
Rocker Bar

Fig. 42.

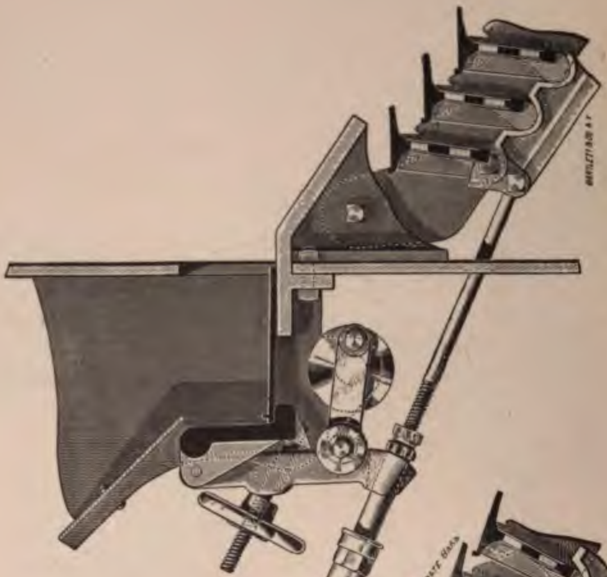

Fig. 43.

on the dumping grate shown by Fig. 44; from time to time the dumping grate may be thrown down as indicated in white lines, and the ashes slide onto the ash-pit floor.

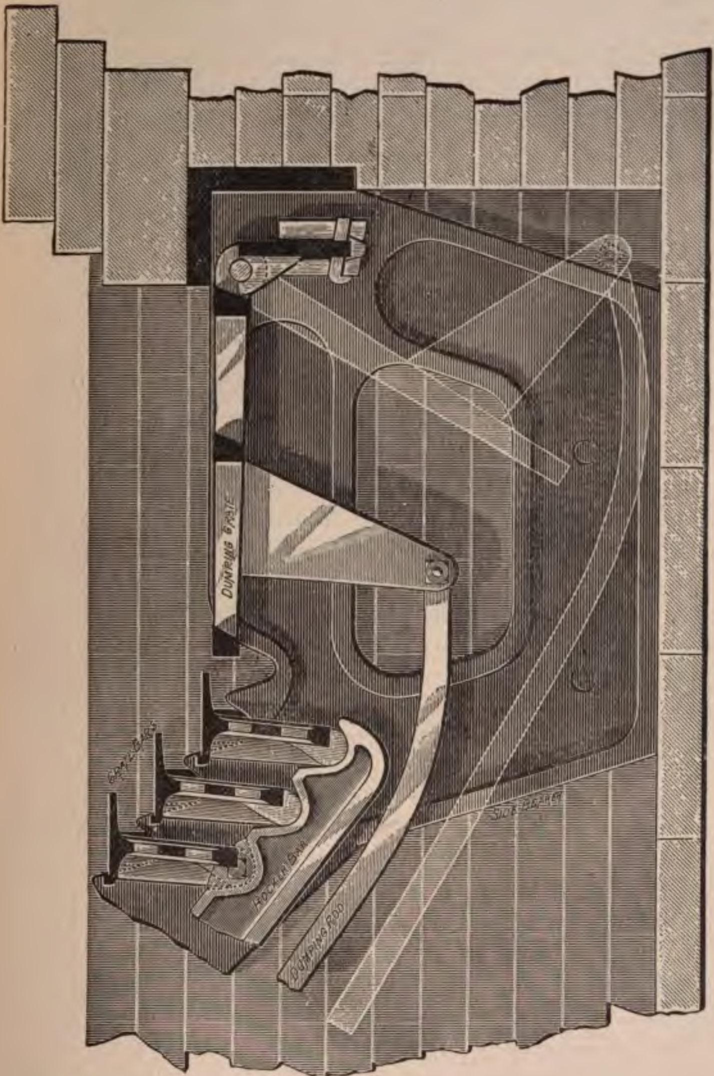

FIG. 44.

This grate appears to be well adapted to burn smoky fuel, as such fuel is well coked at the top of the grate, and the volatile parts driven off by coking can burn with the excess of air coming through the grate at the bottom.

If the rate of feed is too fast it is evident that unburned coal will work down onto the dumping grate, and will appear in the ashes. If the rate of fuel is regulated so that no coal appears in the ashes, the fire becomes thin at the bottom, and an excess of air is liable to enter there; certain tests on this grate have indicated such an excess of air, which is the side on which the fireman is liable to err, as he may not know how much waste he thus occasions, while he can see the coal in the ashes.

Special Furnaces are required for burning various refuse material, such as sawdust, tan bark, straw, and bagasse; no attempt will be made to describe them here.

When wood is burned on a grate it may be sawn into pieces two feet long, and the grate may have the bars spaced wider than for coal. Cord wood can be burned on a brick hearth a little longer than the sticks of wood; the wood ashes are small in amount, and are light, so that the draught will sweep a large portion of them into a pit beyond the hearth. Wood is not now burned for making steam except in remote places, unless it be at sawmills and wood-working factories, where a large amount of refuse wood is produced in the form of slabs, sawdust, shavings, etc.

Smoke Prevention has become a matter of great social importance in cities where much smoky coal is used. Though the loss through imperfect combustion of carbon to the form of carbon monoxide may be great, and though there may be an appreciable loss if the volatile parts of coal are driven off unconsumed, it is a fact that the loss in smoke, even when it is dense and black, is not enough to induce coal users to take the trouble to prevent the formation of smoke. Not infrequently it has been found that the methods used to prevent smoke are accompanied by a loss instead of a gain. For example, smoke burning by the alternate firing of two furnaces leading to a common combustion chamber may give a slightly greater efficiency if just enough hot air in excess is admitted through the clear fire to burn the gases distilled from the fresh charge. If the clear fire must be kept too thin, and thus admit a large amount of air, in order that the smoke may be burned, there will be a loss of efficiency. Though it is not well proved, it is asserted that the mixture of finely divided carbon, in the form of smoke, with carbon dioxide may give a clear gas with the formation of carbon monoxide, and thus with a notable loss. The same difficulties arise when side firing and coking are resorted to with smoky fuels.

The most perfect arrangement for smoke prevention which has yet been tried consisted of a detached furnace with small grate area and a deficient air supply, so that the coal was distilled and burned to carbon monoxide; the resulting hot gases were then burned under a steam boiler. The method was suggested by the producer furnaces, used for making gas for the open hearth in steel making. The objections are the loss of heat by radiation from the detached furnace, and the space occupied by that furnace. Though reported to be a success so far as the prevention of smoke is concerned, it has not met with approval.

Thus far, when laws against making smoke have been enforced, the users of fuel have chosen to buy anthracite coal or coke, or in some cases have used crude petroleum oil.

Oil-Burning Furnaces have the oil thrown in by sprayers or atomizers, and the oil burns in a flame that is about four inches in diameter and two to four feet long. The sprayer has two conical converging tubes, one inside the other, something like the steam and water nozzles of a steam injector. Compressed air or superheated steam is supplied to the inner tube and the oil is drawn through the outer tube and thrown into fine spray mingled with air. Compressed air is the better, considering proper combustion only, but the great convenience of using steam near a steam boiler has led to its common use.

It is very important that the supply of oil should be below the sprayers; it is usually supplied by a small direct-acting steam pump. This is to avoid the accidental flooding of the furnace and fire room with oil and the attendant danger of conflagration.

Forced Draught. — When a higher rate of combustion is required than can be had with natural draught, resort is had to forced draught, by aid of which 100 pounds of coal can be burned per square foot of grate surface per hour.

Three systems of forced draught are in common use, namely, with a *closed stoke hole*, with *closed ash pits*, and *induced draught.*

Induced draught has long been used on locomotives, through the action of the exhaust steam thrown through the smoke stack. The same method is used to some extent on tugboats. This method is simple and effective, but can be used only with non-condensing engines. Attempts have been made to get induced draught by placing a centrifugal or other form of blower in the chimney, but the heat, smoke, and cinders quickly put it out of order.

On steamships forced draught has been obtained by the aid of centrifugal fan blowers. The method with closed ash pit has been used with success on merchant steamers and some war ships. With this method the air drawn from the fire room passes through a blower and is delivered to the ash pit, which has an air-tight door. If the pressure in the ash pit exceeds the resistance to the passage of air through the fuel, flame comes out around the fire door unless it is also made air tight. When the fire door is opened to throw on coal, the blast must be shut off from that furnace and all others having a common combustion chamber, or flame will shoot out into the fire room in a dangerous manner. One reason why it has not been used on war ships is the difficulty of properly ventilating the many small fire rooms in which boilers are placed.

The closed stoke hole has been the customary way of getting a forced draught on torpedo boats and on other naval vessels. The stoke hole is closed air tight, admission and egress being through air locks, and air from without is forced in through a centrifugal blower till the pressure exceeds that of the atmosphere. When a fire door is opened to attend to the fire, there is a strong inrush of air that is liable to make the tube plates leak. So great difficulty has been experienced from this cause, when forced draught has been used with the Scotch boiler, that many naval officers doubt its advisability for large ships. The success of forced draught on the locomotive and on torpedo boats with modified locomotive boilers may be attributed partly to the type of the boiler and partly to the fact that there is only one boiler and one furnace. When two boilers are used on a torpedo boat, each has its own chimney.

On locomotives, the induced draught frequently amounts to a difference of pressure equivalent to that of a column of water five or seven inches high. Forced draught on torpedo boats has approached those figures, but is usually less. Large ships usually have the forced draught restricted to two inches of water. On account of the resistance to the entrance of air to the fire room of war ships, through ventilating shafts, gratings, etc., it has been common to assist the draught by running the blowers without closing the air locks.

Howden's System. — The temperature of gases in the uptakes of marine boilers is frequently high, especially when forced draught is used. In Howden's system, the products of combustion pass among horizontal transverse tubes placed in an enlargement of the uptake. Air to supply the fire is drawn through these tubes by a fan blower and is thereby warmed, thus saving heat and giving quicker combustion. Care must be taken in using this system not to go too far, or the fire may become too hot and rapidly burn out the fire grates and do other injury.

Clearing Fires. — Three tools are used in clearing the grate; they are a long straight bar known as the slice bar, a similar bar with the point bent at right angles to make a hook, and a long-handled rake with three or four prongs. The hook may be run along between the grate bars from below, to clear the spaces from ashes and clinker. The slice bar is thrust under the fire on top of the grate to break up the cinder; it is used also to stir and break up caking coals. The rake is used to haul the fire forward or to draw out cinder.

To clean a fire the fireman breaks up the cinder with the slice bar and rattles down the ashes; if necessary, he works the fire back toward the bridge and exposes the grate in front, which may then be thoroughly cleaned. Then he hauls the fire forward and cleans the back end of the furnace. Cinder which will not break up and pass through the grate is pulled out through the fire door. Some firemen efer to clean the grate one side at a time. After the grate is cleaned e fuel left is spread evenly over the grate and fresh fuel is thrown ı. The fire should be allowed to burn down before cleaning, but fair amount of glowing coal should be left to start a new fire briskly. efore beginning to clean the fire the draught should be checked by osing dampers or otherwise.

CHAPTER VI.

GRATE SURFACE AND HEATING SURFACE.

The Rate of Combustion of fuel is stated in pounds burned per square foot per hour. The following table gives rates of combustion for several types of boilers: —

RATES OF COMBUSTION.

	Pounds per sq. ft. per hour.
With chimney draught:	
Slowest rate, Cornish boilers	4 to 6
Ordinary rate, Cornish boilers	10 to 15
Ordinary rate, factory boilers	12 to 18
Ordinary rate, marine boilers	15 to 25
Quickest rate for anthracite coal	15 to 20
Quickest rate for bituminous coal	20 to 25
With forced draught:	
Locomotives	40 to 100
Torpedo boats	60 to 125

The variation of the rate in any given case depends on the draught, the kind of coal, and the skill of the fireman.

Ratio of Grate Surface to Heating Surface for several types of boilers is given in the following table: —

RATIO OF GRATE SURFACE TO HEATING SURFACE.

Marine return-tube boilers	1 : 25 to 1 : 38
Marine return-tube boilers, average of a large number of boilers	1 : 30
Lancashire boilers	1 : 26 to 1 : 33
Cornish boilers	1 : 27 to 1 : 34
Boilers of modified locomotive type	1 : 30 to 1 : 34
Yacht boilers, locomotive type	1 : 40 to 1 : 46
Horizontal internally fired cylindrical multitubular boilers	1 : 45 to 1 : 50
Portable boilers, locomotive type	1 : 23 to 1 : 70
Water-tube boilers	1 : 34 to 1 : 65
Locomotive boilers	1 : 60 to 1 : 70

In calculating the heating surface of a given boiler, it is customary to take account of all the surface that comes in contact with the hot gases. Thus for a cylindrical tubular boiler we include that part of the cylindrical shell which is below the supports on the side walls, the rear tube plate up to the brick arch that protects the steam space from the hot gases, and all the inside surface of the tubes. The area of the tubes is deducted from the gross area of the tube plate, to find the effective area. The front tube plate is not counted as heating surface. The heating surface of a locomotive boiler consists of the four sides and crown of the fire box and the inside of the tubes; allowance is of course made for the area of the tubes in calculating the area of the front side of the fire box, *i.e.* of the tube plate.

Relative Value of Heating Surface. — It is apparent that there is much difference in the value of the different parts of the heating surface depending on its form and location; consequently when we use the above table or any other tables, it must be with the understanding that they apply to boilers of ordinary dimensions and proportions.

Hutton[1] gives the following estimate of the relative values of heating surface from certain parts of some types of boilers; even if not entirely reliable, such an estimate gives a good general idea of the relative values: —

Crown sheet of locomotive boiler	1.00
Crown of furnace in flue	0.95
Plates of cylindrical boiler over external furnace	0.90
Fire-box tube plate of locomotive boiler	0.80
Water-tube surface, facing fire	0.70
Vertical side of fire box	0.50
First three feet of tube of locomotive boiler	0.30

Surfaces below the fire grate are not counted as heating surface; for example, the part of the flue under the grate in a Lancashire boiler.

He also gives as the results of four experiments to determine the evaporative power of the heating surface of the plates of a locomotive fire box, —

19.40 20.64 21.96 23.12

average, 21 pounds per square foot per hour. He quotes 16.67 pounds for one experiment and 36.9 for another, on other boilers. Experiments on two locomotive boilers gave 6.8 and 9.6 pounds of water evaporated per square foot of tube surface per hour.

In an experiment with a multitubular boiler, the tubes were divided into six sections, and the percentage of evaporation by each section was as follows: —

Sections of tubes	1,	2,	3,	4,	5,	6.
Percentage of evaporation	47,	23,	14,	8,	5,	3.

In another experiment with a multitubular boiler, the barrel was divided by partitions into five lengths, each 12 inches long; the percentages of evaporation from the different divisions were as follows: —

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[1] Steam-boiler Construction.

Section of tubes	1,	2,	3,	4,	5.
Percentage of evaporation	65.	29,	16,	13,	10.

In both of these tests the high percentage of the first section is due largely to the aid of the tube plate. The diameters of the tubes were probably about 2 inches, but it is not stated by Hutton.

It is probable that there is little if any advantage in making two-inch tubes more than 8 feet long for horizontal boilers; for vertical boilers they may be twice as long with advantage. The four-inch tubes for horizontal stationary boilers, in like manner, need not be more than 16 feet long. If it is desired that flame shall traverse a flue without being extinguished, the flue should not be less than one foot in diameter.

Equivalent Evaporation. — The heat required to evaporate one pound of water depends on the temperature of the feed water, the pressure of the steam, and the per cent of priming. Thus to evaporate a pound of water from the temperature t_0° F. under the pressure p in pounds per square inch absolute, with $1 - x$ priming (*i.e.* with a quality of steam x) requires

$$xr + q - q_0$$

thermal units. Here r is the heat of vaporization, and q the heat of the liquid corresponding to the pressure p, and q_0 is the heat of the liquid at the temperature t_0. Now the heat of vaporization at 212° is 965.8 thermal units, and

$$\frac{xr + q - q_0}{965.8}$$

is called the equivalent evaporation from and at 212°. This is equivalent to using a special thermal unit 965.8 times as large as the standard unit.

Evaporation per Pound of Coal depends on the quality of the coal, the rate of combustion, and the type of the boiler. The influence of the temperature of the feed water and the pressure of the steam is commonly allowed for by giving the equivalent evaporation at 212°.

It is usual also to state the evaporation per pound of combustible, allowing for the ashes and other incombustible refuse. It is, however, to be remembered that the carbon in a poor fuel has a less value than an equal quantity of carbon in a good fuel; so that a good fuel should always be chosen for making comparative tests of different boilers, unless the use of poor fuel is an essential element of the test.

The effect of the rate of combustion is illustrated by the following table given by Isherwood as the result of tests made on a marine tubular boiler which had a ratio of heating surface to grate surface of 25 to 1. The equivalent evaporation at 212° is given in this table.

Rate of Combustion, Pounds of Coal per Square Foot of Grate Surface.	Evaporation per Pound of Coal.	Evaporation per Square Foot of Heating Surface.
6	10.5	2.52
8	10.4	3.33
10	10.1	4.04
12	9.5	4.56
14	8.9	4.98
16	8.2	5.25
18	7.7	5.54
20	7.3	5.84
22	7.0	6.16
24	6.8	6.53

As a matter of interest, the evaporation per square foot of heating surface has been added; it is obtained by multiplying the evaporation per pound of coal by the rate of combustion, and then dividing by the ratio of heating to grate surface. The greater efficiency of the heating surface is due to the stronger draught required for the higher rate of combustion, and to the resulting increase of the temperature of the fire and of the hot gases in the tubes.

Hutton gives the following table of equivalent evaporation per pound of coal for several types of boilers: —

Plain cylindrical	5 to 8
Vertical	5 to 10
Water-tube	5 to 11
Cornish	6 to 11
Lancashire	6½ to 12
Marine return tube	7 to 12
Multitubular	8 to 12
Galloway	9 to 12½
Locomotive	8 to 13

The lower rate is what may be expected with the boiler in ordinary condition and with ordinary care in firing. The superior limit is the highest that can be expected with good design, clean plates and tubes, slow rate of combustion, and extra care and skill; results as high as 12 pounds, and over, per pound of coal should receive credence only when the tests are made by competent observers under favorable conditions.

For cylindrical tubular boilers, and similar boilers, under ordinary conditions, an actual evaporation of 8½ pounds of water, or an equivalent evaporation of nine pounds of water, per pound of coal is all that can be expected.

Locomotive boilers, on railroads, more commonly give six or seven pounds of water per pound of coal.

The vertical boilers quoted are small portable boilers; large, well-designed, vertical boilers, with tubes two inches in diameter and 16 feet long, give as good results as any multitubular boilers.

As an example of high economy, we may quote the official test on the boilers at the Boston Main Drainage Pumping Station.[1] These boilers are of the locomotive type, with large double furnaces, and with combustion chambers in the barrel between the furnaces and the tube plates.

The following table gives the data and results of a test on a boiler of similar construction, designed by Mr. F. W. Dean, for the Newton Water Works: —

TEST OF BOILER AT NEWTON UPPER FALLS.

Duration of test, 11 hours 45 minutes, or	11.75	hours.
Average pressure, by gauge	122.87	pounds.
Atmospheric pressure	14.8	pounds.
Average absolute pressure	137.67	pounds.
Average temperature of feed	116°.06	F.
Total heat above 32° F., of steam of 137.67 pounds per sq. in.	1189.1	
Heat in water of temperature of 116°.06 F., above 32° F.	84.1	
Heat given to every pound of water fed to the boiler, under the assumption that it was evaporated into dry and saturated steam of the boiler pressure, 1189.1 – 84.1 =	1105.0	B. T. U.
Total water fed to the boiler	44704.5	pounds.
Total coal consumed	4548.1	pounds.
Water evaporated per pound of coal at boiler pressure and temperature of the feed, 44704.5 ÷ 4548.1 =	9.83	pounds.
Ash per pound of coal	0.057	pound.
Combustible per pound of coal	0.943	pound.
Water evaporated per pound of combustible at boiler pressure and temperature of feed = 9.83 ÷ 0.943 =	10.42	pounds.
B. T. U., per pound of coal, given to every pound of water fed to the boiler, under the assumption that it was evaporated into dry and saturated steam of the boiler pressure, 9.83 × 1105 =	10862.15	
Equivalent evaporation per pound of coal from and at 212° F., = 1105 × 9.83 ÷ 965.8 =	11.25	pounds.
Equivalent evaporation per pound of combustible from and at 212° F., 11.25 ÷ 0.943 =	11.93	pounds.
Area of grate	48¼	sq. ft.
Coal burned per sq. foot of grate per hour = $\frac{4548.1}{11.75 \times 48\frac{1}{4}}$ =	8.04	pounds.

Grate Surface and Heating Surface per Horse Power. — On account of the difficulty of making boiler tests on steamships, and the consequent lack of exact information, it is customary to base the design of marine boilers on the indicated horse power of the engines, which can be readily determined. The following table from Hutton represents English practice; a few land boilers are included for comparison.

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[1] Thermodynamics of the Steam Engine. — PEABODY. Page 297.

GRATE SURFACE AND HEATING SURFACE PER HORSE POWER.

	Square Feet Grate Surface.	Square Feet Heating Surface.
Boilers of locomotive engines	0.021 to 0.034	1.4 to 1.85
Boilers of yachts, locomotive type	0.035 to 0.040	1.55
Marine boilers, modified locomotive type	0.050 to 0.070	1.6 to 2.0
Return-tube boilers of armored ships	0.050 to 0.070	
Coil boilers	0.050 to 0.083	1.5 to 2.25
Return-tube boilers for express steamers	0.083	
Return-tube boilers, dispatch boats	0.065 to 0.095	
Marine return-tube boilers for triple engines	0.080 to 0.100	3.30 to 3.60
Torpedo-boat boilers, locomotive type	0.100 to 0.125	1.33
Marine return-tube boilers, compound engines	0.090 to 0.140	4.00
Marine return-tube boilers, average of a large number of boilers	0.120	3.25
Lancashire boilers	0.100 to 0.165	2.75 to 4.25

All these boilers have natural draught, except the locomotive-engine boilers and the yacht-locomotive boilers, which have exhaust steam blast in the chimney. The heating surface per horse power for marine return-tube boilers ranges from 1.75 to 2.5 square feet, and may, in a general way, be averaged at two square feet.

Mr. Byles estimates that warships during continuous steaming will develop one horse power for each three feet of heating surface, and on this basis estimates the sustained sea speed of the U. S. S. *Columbia* at 20½ knots: the average speed during the four hours of the trial trip was 22.8 knots.

Steam Space. — The steam space and the free surface for the disengagement of steam should be sufficient to provide for the efficient separation of the steam from the water. Cylindrical-tubular boilers frequently have the steam space equal to one-third of the gross volume of the boiler shell. Marine return-tube boilers usually have a smaller ratio of steam space to water space.

The more logical way appears to be to proportion the steam space to the rate of steam consumption by the engine. Thus the ratio of the volume of the steam space of cylindrical boilers to that of the high-pressure cylinder of multiple expansion engines varies from 50:1 to 140:1. The ratio of the steam space of a simple locomotive engine to the volume of the two cylinders is about 6½:1.

The capacity of the steam space is sometimes equal to the volume of steam consumed by the engine in 20 seconds. It was found in some experiments with marine boilers, having a working pressure less than 50 pounds per square inch, that a considerable quantity of water was carried away by the steam when the steam space was equal to the volume of steam consumed in 12 seconds, but that no water was carried into the cylinders when the steam space was equal to the volume of steam used in 15 seconds, and that no trouble from water was ever experienced when the steam space was proportioned for 20 seconds.

All the preceding discussion refers to engines that run at a considerable speed of rotation; not less than 60 revolutions per minute. Engines that make but few revolutions per minute and take steam for only a portion of the stroke, require a larger proportion of steam space. As an example, we may cite the walking-beam engines for paddle steamers.

Boiler Horse Power. — Since the power from steam is developed in the engine, and not in the boiler, and since the economy of the use of steam depends on the engine only, it is apparent that we cannot properly assign any horse power to a boiler. Still it is customary and convenient to rate boilers by the horse power, and for that purpose it is customary to consider that 30 pounds of water evaporated from feed water at 100° F., under the pressure of 70 pounds by the gauge, is equivalent to one horse power. This is the standard recommended by a committee of the American Society of Mechanical Engineers.[1]

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[1] Trans., Vol. vi., 1881.

CHAPTER VII.

CHIMNEYS AND FLUES.

At the present time, the theory of chimney draught is in such an unsatisfactory condition, on account of the lack of proper experimental investigations, that in practice flues and chimneys are proportioned empirically or by comparison with examples that give good results. Common proportions will be given first, and then a statement of the present condition of the theories.

Height of Chimney. — Professor Trowbridge gives the following table of heights of chimney required to give certain rates of combustion, obtained by collecting reliable data and drawing a curve to represent mean results: —

HEIGHT OF CHIMNEY.

Heights in Feet.	Pounds of Coal per Square Foot of Section of Chimney per Hour.	Pounds of Coal per Square Foot of Grate per Hour.
20	60	7.5
25	68	8.5
30	76	9.5
35	84	10.6
40	93	11.6
50	105	13.1
60	116	14.6
70	126	16.3
80	135	[illegible]
90	144	[illegible]
100	152	[illegible]
110	160	[illegible]

Area of Section. — The area of the section of the chimney may be inferred from the rate of combustion per square foot of the section of the chimney, in the preceding table. For marine boilers with chimneys of circular section with smooth internal surfaces, and [illegible] unobstructed flues, we may have the following proportions [illegible] compared with the grate area: —

Section of chimney	[illegible]
Section of tubes, total	[illegible]
Area over fire bridge	[illegible]

Chimney Draught. — The density of the chimney gases is slightly greater than that of air at the same temperature and pressure, but, for our present purpose, we may neglect the difference, and proceed as though we were dealing with air only.

Let v_0 be the specific volume of air at the normal pressure of the atmosphere, and at the temperature of freezing water, which will be represented by T_0 on the absolute scale. Let T_a and T_c be the absolute temperatures of the atmosphere in the chimney. Then, neglecting changes of atmospheric pressure, we shall have for the weights of one cubic foot of the atmosphere, and of the chimney gas,

$$\frac{1}{v_0}\frac{T_0}{T_a}, \qquad \frac{1}{v_0}\frac{T_0}{T_c}. \tag{1}$$

The weights of a column of air H feet high and one square foot in area of cross-section, under the two conditions, will be

$$\frac{H}{v_0}\frac{T_0}{T_a}, \qquad \frac{H}{v_0}\frac{T_0}{T_c},$$

and the difference of the weights will be

$$\frac{H}{v_0}\left(\frac{T_0}{T_a}-\frac{T_0}{T_c}\right). \tag{2}$$

If we have the column of hot air at rest in a chimney, the difference, at the foot of the chimney, between the pressure inside and outside will be represented by the expression (2), in pounds to the square foot. If we wish to express the pressure in inches of water, we will multiply by $\frac{12}{62.4}$; this method is convenient when the draught is measured by a U-tube containing water.

All theories of chimney draught that have been proposed treat the difference of pressures, inside and outside the chimney, as though it were a head producing a flow of a fluid, as a head of water produces a flow of that liquid.

Flow of a Liquid. — In hydraulics, it is shown that we may express the relation between the velocity of flow of a liquid in a pipe and the head producing the flow, by the following equation: —

$$h=\frac{V^2}{2g}\left(1+k+k_1+\frac{fl}{m}\right), \tag{3}$$

in which h is the head, in feet, of the liquid producing the flow; V is the velocity in feet per second; g is the acceleration due to gravity; k and k_1 are coefficients used to express the resistance of obstructions like valves, bends, etc.; l is the length of the pipe; m is the ratio of the area to the perimeter of the pipe; and f is the coefficient of friction of the fluid against the sides of the pipe. The several coefficients vary with the velocity of flow, the size of the pipe, and the nature of the resistance. Experiments in hydraulics have been made, and tables prepared, so that the proper coefficients may be selected for use with the equation under any given conditions.

It is assumed in theories of chimney draught that an equation of the same form may be used to express the relation between the head, or difference of pressure inside and outside the chimney, and the flow of gases through the furnace, flues, and chimney. The resistances to the passage of gases through the grate and the fuel on it, and through the flues and tubes, may be expressed by aid of coefficients G and C, which are like k and k_1 in equation (3); the resistance of friction of the gases against the side of the chimney may be assimilated to the last term in the parenthesis of equation (3), replacing l by H.

Peclet's First Theory. — The theory of chimney gases commonly given in text-books and elsewhere was proposed by Peclet several years ago. In this theory he expressed the head producing the flow in feet of hot gas in the chimney, so that from expression (2) we have

$$h = \frac{H}{v}\left(\frac{T_0}{T_a} - \frac{T_0}{T_c}\right) \div \frac{1}{v}\frac{T_0}{T_c} = H\left(\frac{T_c}{T_a} - 1\right). \qquad (4)$$

He then takes for his equation expressing the relation of the height of the chimney and the velocity V_c of the hot gases therein

$$\frac{V_c^2}{2g}\left(1 + G + C + \frac{fH}{m}\right) = H\left(\frac{T_c}{T_a} - 1\right).$$

$$\therefore V_c = \sqrt{2gH}\left(\frac{T_c}{T_a} - 1\right)^{\frac{1}{2}} \frac{1}{\left(1 + G + C + \frac{fH}{m}\right)^{\frac{1}{2}}}. \qquad (5)$$

If the area of cross-section of the chimney is A square feet, the volume of hot gas discharged per second is

$$V_cA,$$

and the weight discharged per second is

$$W = V_cA \cdot \frac{1}{v}\frac{T_0}{T_c}.$$

$$\therefore W = \frac{A}{v}\sqrt{2gH}\,\frac{T_0}{T_c}\left(\frac{T_c - T_a}{T_a}\right)^{\frac{1}{2}} \frac{1}{\left(1 + G + C + \frac{fH}{m}\right)^{\frac{1}{2}}}. \qquad (6)$$

From some experiments on chimneys and boilers, Peclet gives, in connection with this first theory, the following values for the coefficients

$$G = 12, \qquad f = 0.012,$$

under the assumption that from 20 to 24 pounds of coal are burned per square foot of grate per hour; the coefficient C does not appear in his equation.

Equation (6) is in the proper form for calculating the weight of gas discharged by a given chimney for which the height, area, and perimeter of cross-section are known. If the weight of gas to be discharged and the area and perimeter are known, the equation for a given case leads to a quadratic equation for finding the height H, which can readily be solved numerically. If the weight of gases is known, and the height of the chimney is assumed, then the insertion of linear dimensions in place of A and m leads to an equation of the fourth degree; but as $\frac{fH}{m}$ is small compared with $1 + G$, an approximate solution may be had by neglecting this term; the second approximation by inserting for m a value deduced from the first approximation to A could be readily obtained if desired.

It appears that the equation (6) gives a maximum for a value of $T_c = 2\,T_a$.

The factor in equation (6), which varies with the temperature is,

$$\frac{(T_c - T_a)^{\frac{1}{2}}}{T_c};$$

equating the first differential coefficient with regard to T_c, to zero we have

$$T_c \tfrac{1}{2}(T_c - T_a)^{-\frac{1}{2}} - (T_c - T_a)^{\frac{1}{2}} = 0.$$

$$\therefore\ T_c = 2\,T_a;$$

that is, the maximum weight of gas discharged by the chimney, according to this theory, will occur when the absolute temperature in the chimney is twice the temperature of the atmosphere.

If the temperature of the atmosphere is 70° F., this hot gas theory gives for the temperature producing the maximum flow

$$2(460 + 70) - 460 = 600^\circ \text{ F.},$$

a figure that is often quoted.

Peclet's Second Theory. — A later theory, developed by Peclet, made the velocity of the gas entering the furnace depend on the difference of pressure inside and outside of the chimney, expressed in feet of cold gas. This gives for the head

$$h = \frac{H}{v}\left(\frac{T_0}{T_a} - \frac{T_0}{T_c}\right) \div \frac{1}{v}\,\frac{T_0}{T_a} = H\left(1 - \frac{T_a}{T_c}\right). \tag{7}$$

In assimilating the flow of gas through the furnace, flues, and chimney, we now have the resistance to the flow through the grate *and* the fuel on it, proportional to the square of the velocity of the gas entering the chimney, while the resistance to the flow through flues and tubes and the resistance due to friction in the chimney, are proportional to the velocity of the hot gases; neglecting variations of temperature in the flues and tubes, we may take for the temperature of the hot gases T_c. Now the specific volume of the cold gases and hot gases are proportional to the absolute temperatures, and the velocities of flow through passages having equal areas of cross-section will be proportional to the specific volumes, so that we may assume the velocity of the hot gases to be

$$V_c = \frac{T_c}{T_a} V_a. \tag{8}$$

Our equation for flow may consequently be written

$$2g V_a^2 \left(1 + G + C\frac{T_c}{T_a} + \frac{fH}{m}\frac{T_c}{T_a}\right) = H\left(1 - \frac{T_a}{T_c}\right).$$

$$\therefore V_a = \sqrt{2gH}\left(1 - \frac{T_a}{T_c}\right)^{\frac{1}{2}} \frac{1}{\left(1 + G + C\frac{T_c}{T_a} + \frac{fH}{m}\frac{T_c}{T_a}\right)^{\frac{1}{2}}}. \tag{9}$$

The weight of hot gas discharged per second is therefore

$$W = V_c A \cdot \frac{1}{v}\frac{T_0}{T_c} = V_a A \frac{1}{v}\frac{T_0}{T_a},$$

from the equation (8). Consequently,

$$W = \frac{A}{v}\sqrt{2gH}\frac{T_0}{T_a}\left(\frac{T_c - T_a}{T_c}\right)^{\frac{1}{2}} \frac{1}{\left(1 + G + C\frac{T_c}{T_a} + \frac{fH}{m}\frac{T_c}{T_a}\right)^{\frac{1}{2}}}. \tag{10}$$

If we assume that the terms $C\frac{T_c}{T_a}$ and $\frac{fH}{m}\frac{T_c}{T_a}$ are small as compared with G, then the weight discharged varies with

$$\left(1 - \frac{T_a}{T_c}\right)^{\frac{1}{2}},$$

and this expression increases without limit as T_c increases. This theory is considered to show that there is no limiting temperature like that shown by the first theory. But, for boilers with fire tubes, the term $C\frac{T_c}{T_a}$ is not small compared with $1 + G$, so that the flow should be considered to vary with

$$\left(\frac{T_c - T_a}{T_c}\right)^{\frac{1}{2}} \frac{1}{\left(1 + G + C\frac{T_c}{T_a}\right)^{\frac{1}{2}}},$$

and this expression will give a maximum when the first differential coefficient is equated to zero.

Discussion of Peclet's Theories. — In the first place, it is evident that the two theories are incompatible; for, neglecting the resistances due to friction and to the passage of the gases through the fuel on the grate, and through the tubes, we have by the two theories

$$V_c = \sqrt{2gH}\left(\frac{T_c - T_a}{T_a}\right)^{\frac{1}{2}}, \qquad V_a = \sqrt{2gH}\left(\frac{T_c - T_a}{T_c}\right)^{\frac{1}{2}},$$

so that

$$\frac{V_c}{V_a} = \left(\frac{T_c}{T_a}\right)^{\frac{1}{2}}.$$

But since the pressure in the chimney is assumed to be constant, [illegible]e should have

$$\frac{V_c}{V_a} = \frac{T_c}{T_a}.$$

In the fourth edition of Peclet's work,[1] — the editor, A. Hueld decides, from a special thermodynamic investigation, that the first o hot gas theory should be chosen. An application of the general theory of the flow of fluids appears to favor the opposite conclusion; both however, take no account of friction and resistances, and cannot be conclusive.

In the general theory of the flow of fluids,[2] it appears that the assumption that the volume of the fluid is not appreciably affected by changes of pressure and temperature leads at once to the equation

$$V = \sqrt{2gh},$$

which is used in hydraulics for calculating the velocity of flow of a liquid from a reservoir. But a gas experiences notable changes of volume from a change of either pressure or temperature; consequently the above equation cannot give accurate results even for slight changes of pressure. Both of Peclet's theories make use of this equation, and are, therefore, equally defective in theory, even though they should have a practical convenience.

If we consider the mechanical assimilation of the flow of hot gas in a chimney to the flow of water in a pipe, the first or hot-gas theory appears preferable, since it expresses the head in terms of the fluid with whose velocity the resistances are supposed to vary. Moreover, nothing but hot gas gets into a chimney, and the velocity of flow of the cold gas approaching the furnace must depend on the area of the conduit leading to the furnace, which may differ from the area of the chimney.

On the other hand, if we try to assimilate the chimney to a pipe discharging air from a reservoir into the atmosphere, the cold gas theory appears to have the advantage. To make the idea more definite, a calculation will be carried through for a special case, omitting all resistances.

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[1] Traité de la Chaleur, etc. — E. Peclet, Tome 1, page 280.

[2] *Thermodynamics* of the Steam Engine, page 129.

Let it be assumed that the difference of pressure between the gases in the chimney and the air outside at the base is equivalent to one inch of water, or 5.2 pounds per square foot. Let the temperature of the atmosphere be 60° F., at which the specific volume is

$$12.39 \times \frac{460.7 + 60}{492.7} = 13.09 \text{ cubic feet.}$$

The thermodynamic equation for the flow of cold air into the chimney, as through a short tube into the air, gives

$$V = \left\{ 2 g p_1 v_1 \frac{\kappa}{\kappa - 1} \left[1 - \left(\frac{p_2}{p_1} \right)^{\frac{\kappa - 1}{\kappa}} \right] \right\}^{\frac{1}{2}}.$$

$$\therefore\ V = \left\{ 2 \times 32.16 \times 14.7 \times 144 \times 13.09 \frac{1.4}{1.4 - 1} \left[1 - \left(\frac{144 \times 14.7 - 5.2}{144 \times 14.7} \right)^{\frac{1.4 - 1}{1.4}} \right] \right\}^{\frac{1}{2}}.$$

$$\therefore\ V = 64.4 \text{ feet per second.}$$

Using the formula borrowed from hydraulics, we have for the head,

$$5.2 \times 13.09 = 68.07 \text{ feet,}$$

and the velocity will apparently be

$$V = \sqrt{2 \times 32.16 \times 68.07} = 66.2 \text{ feet.}$$

The discrepancy is something less than three per cent. This discrepancy will be less for smaller differences of pressure, and more for larger differences; but the variation is not much for any case arising in practice: for example, the discrepancy is about eight per cent for 4.7 pounds difference, equivalent to 130 inches of water.

From Fliegner's experiments it appears that the pressure in the orifice is greater than the pressure of the external atmosphere where air discharges from a reservoir in which the pressure is much greater than that of the atmosphere. But for small differences of pressure, the pressure in the orifice approaches the pressure of the external atmosphere. We may, therefore, ignore this matter in comparing the results by the two methods of calculation.

If, now, we may assume that the air once in the chimney is in some way strongly heated without disturbing the flow into the chimney, we may make the velocity of the hot gas leaving the chimney proportional to its specific volume. Ignoring the slight changes of temperature and volume due to the adiabatic flow of air into the chimney, we will assume that the absolute temperature in the chimney is twice the absolute pressure of the atmosphere, or that

$$T_c = 2\,T_a = 2 \times 520.7 = 1041^\circ.4;\ \textit{i.e.}\ 580^\circ\ \text{F.};$$

and we will further assume that the volume of the hot gas is twice the specific volume of the cold gas, or

$$2 \times 13.09 = 26.18 \text{ cubic feet;}$$

so that the velocity of the hot gas is

$$2 \times 64.4 = 128.8 \text{ feet per second;}$$

or if we use the hydraulic equation,

$$2 \times 66.2 = 133.2 \text{ feet per second.}$$

The hot-gas theory will have for the head producing the flow

$$5.2 \times 26.18 = 136.14 \text{ feet.}$$

The corresponding velocity of flow will be

$$\sqrt{2 \times 32.16 \times 136.14} = 93.6 \text{ feet.}$$

This is of course equal to 66.2 multiplied by $\sqrt{2}$, because the specific volume of the hot gas has been assumed to be twice that of the cold gas.

There does not seem to be any *a priori* way of deciding which result is the more probably correct, and as yet no experiments have been made which determine the velocity of gas in a chimney due to given draught. We do not even know whether the method of calculating the draught from the temperatures of the atmosphere and of the chimney gases is justifiable.

Finally, the resistances to the passage of air through the fuel and through the tubes are so much greater than the head required to produce the velocity of flow, that any uncertainty in determining the latter will cause wide discrepancies in the experimental determination of the corresponding constants C and G. It is more than doubtful if so large coefficients of correction for resistance are admissible, especially as neither the temperature nor the velocity of the gases at the grate and at the tubes is anywhere near those for either the hot or the cold gas.

Before a working theory can be produced, we ought to find experimentally —

(1) What draught will be produced by a given height of chimney with a given difference of temperatures inside and outside of the chimney.

(2) What portions of the draught are used in forcing the air through the fuel, and through the flues and tubes, etc., leading to the chimney.

These experimental data should be obtained for a sufficient variety of conditions, including different types and proportions of boilers and various rates of combustion for different kinds of fuel. The time has gone by when engineers expect to get valuable results from theories that are developed by questionable methods, and which depend on a few isolated experiments.

Forms of Chimneys. — Chimneys are commonly built of brick, and are either round or square in section; the round section gives less resistance of friction and is less affected by pressure of wind tending to blow it down. Chimneys are also made of plate iron, and may then be lined with brick to reduce loss of heat.

Figure 45 shows the elevation and section of a large brick chimney; it has a brick lining, separate from the external shell, to provide for expansion. Figure 46 shows the construction of the head

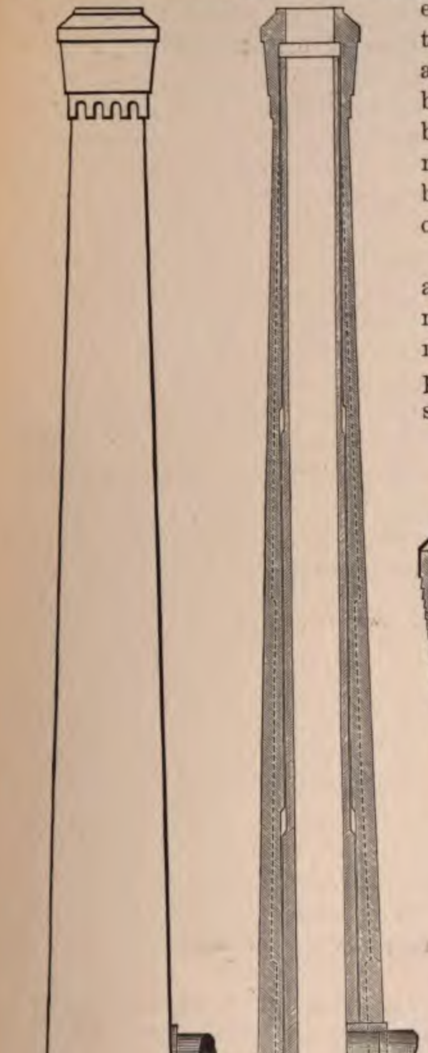

FIG. 45.

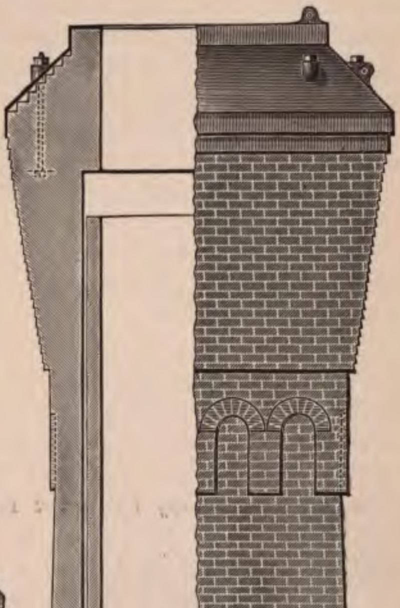

FIG. 46.

of the chimney, which has a cast-iron cap to protect the masonry from infiltration of water. These figures represent a square chimney, but with a few minor changes may be made to represent a round chimney.

Stability of Chimneys. — On account of the concentration of weight on a small area, and the disastrous results that would follow from defective work, the foundations of an important chimney should be carefully laid by an experienced engineer. A natural foundation is to be preferred, but piling and other artificial methods of preparing the earth for the foundation can be used when necessary. Good natural earth should carry from 2000 to 4000 pounds to the square foot. The base of the chimney should be spread out so that this pressure, or whatever the earth can safely bear, may not be exceeded.

A discussion of wind pressure will be found on page 152 of Lanza's *Applied Mechanics*, with a statement of experimental results. Experience shows that an allowance of 55 pounds for wind pressure on flat surfaces of chimneys is sufficient: this figure is higher than that obtained by experiments. The pressure on a round chimney may be taken to be half that on a square chimney, having its side equal to the diameter of the round chimney. The chimney may be treated as a cantilever uniformly loaded with 55 pounds to the square inch, and the bending moments and the resulting stresses should be calculated for sections at intervals from the top down to the base. The stress from the wind pressure is to be compounded with the direct stress due to the weight of the chimney above the section in question. The combined stresses on the side toward the wind must never be a tension. The total compressive stress on the side away from the wind should never be greater than brick masonry can safely bear, say eight tons to the square foot. Only the outer shell should be considered in these calculations. The lining may be calculated for weight only.

Iron chimneys may be calculated in a similar manner, but the shell of such a chimney may endure tension as well as compression. The brick lining, when one is used, adds nothing to the strength of the chimney, but, on the other hand, the lining usually is carried by the shell. The calculations should be made for the weakest sections, which will probably be at the ring seams. If the ring seams are carefully butted and strapped, the compression may be direct from one plate to the next; such seams are liable to be held by the riveting in compression as well as in tension. It is customary to consider that, as well-driven rivets fill the rivet holes completely, the whole section of the plate may be used in calculations for compression. It is better to take the net section, subtracting the rivet holes for compression as well as for tension. Iron chimneys are commonly stayed; in such cases proper allowance should be made in the calculations.

CHAPTER VIII.

STAYING.

ALL sheets of a boiler that are not either cylindrical or hemispherical have to be stayed to keep them in shape. Take, for example, the common form of horizontal multitubular boiler: the sheet is cylindrical and requires no staying, as the internal pressure tends to keep it cylindrical; but the heads, when subjected to pressure, would tend to become hemispherical, and this tendency has to be resisted by the staying.

Different forms and arrangements of staying have to be used with different types of boilers. A few general methods can be given, but the details and final arrangement should be laid out for each boiler.

Horizontal Multitubular Boilers. — In staying a flat surface, it is first necessary to find how near together the stay rivets must be in order that the plate between these rivets shall not have an outside fibre stress of over 6000 pounds per square inch for iron, or 8000 pounds per square inch for steel. The pitch of these stay rivets will generally be from 3.5 inches to 5.0 inches, depending, of course, on the thickness of plate, pressure, etc.

As it is not advisible to run a brace or through-stay from each stay rivet, a number of these are fastened to a channel bar or angle iron, which acts as a beam; this beam being supported at two or three points by means of tie rods or braces. This beam must of course be strong enough to carry the loads brought to it by the rivets, and the tie rods or braces which act as the supports for the beam must be strong enough to carry the entire load on it. Plate I. gives a good example of a form of staying much used to-day. The load on the stay rivets is taken by two six-inch channel bars and by an angle-iron stiffener near the top of the head.

The lower channel is really a continuous girder of four spans and five supports. Three of these supports are the through-stay rods.

The other two supports are at the inner edge of the flange of the head. The upper channel has but two through-stay rods located sixteen inches on centres. These are kept apart so as to enable a man to get into the boiler through the manhole, which would of course be impossible if the arrangement was the same as in the lower channel.

It is not a good plan to stay heads so as to make them perfectly rigid, as this is apt to cause grooving. A space of at least three inches is left between the top of the tubes and the first row of stay rivets, and a similar space is left at the sides. After deciding on the type of staying to be used, the pitch of stay rivets may be made smaller than the calculated amount if it is more convenient in spacing. The vertical pitch is often made different from the horizontal, for the same reason.

The stay rods are upset at the ends so that the diameter at the bottom of the screw thread is as great as the diameter at the middle of the rod. A check nut and washer are placed on the inside, and a copper washer and cap nut on the outside (see enlarged drawing, Plate I.). This copper washer is cupped out on the under side so that it bears on the plate along a narrow ring. The cap nut prevents water from leaking out around the screw thread. Long stay rods, large in diameter, are often supported in the centre.

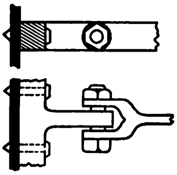

FIG. 47.

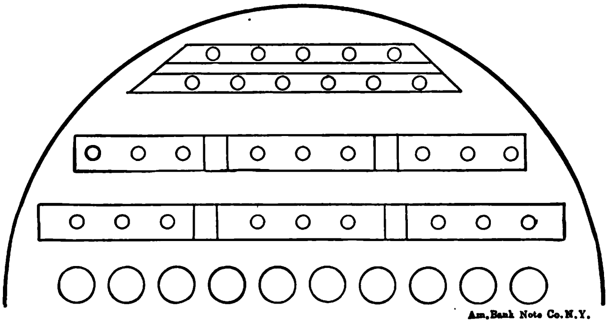

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FIG. 48.

In a similar form of staying often used on horizontal multitubular boilers the channel bar is replaced by two angle irons. The inner washer then bears directly against the plate; other details are the same.

A third style formerly used is shown in Figs. 47 and 48.

Horizontal rows of stay rivets are fastened to bars of wrought iron 2.0 to 2.5 inches square. Ears are jump welded onto these bars, and the stay rods having forked ends are attached to the ears by bolts or split pins. The top of the head is stiffened with angle irons.

A fourth style, *i.e.* the crowfoot, is often used by itself and also in combination with some of the previous methods. An example of crowfoot staying is shown in Fig. 50. It will be noticed that the lower stay rivets in the middle crowfoot are placed too near the top row of tubes. A double crowfoot covers four stay rivets (Fig. 51), while a *single crowfoot* (Fig. 49) takes the load on two rivets. This load is generally carried to the shell by a diagonal brace. Single crowfeet are sometimes made out of pieces of boiler plate bent into the right shape.

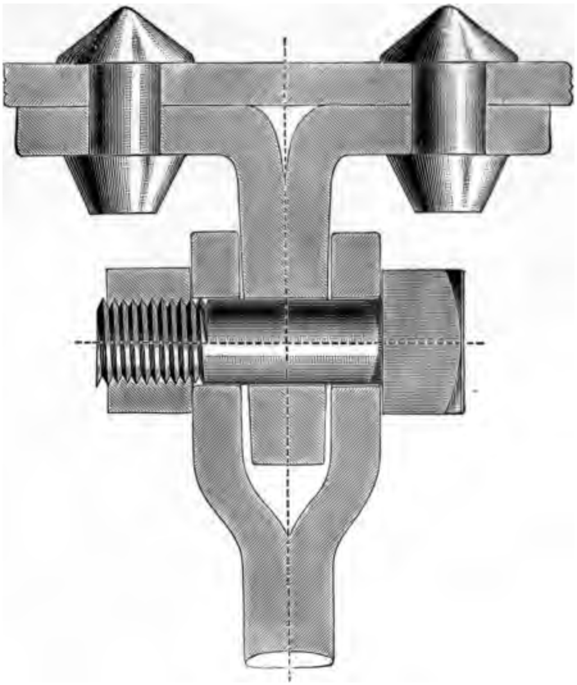

Fig. 49.

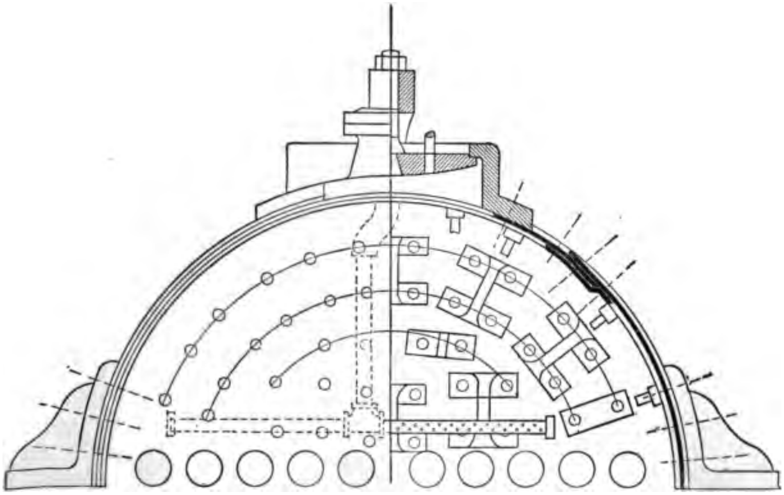

Fig. 50.

Two other forms are shown by Figs. 52 and 53, and the braces used with these forms by Figs. 54 and 55.

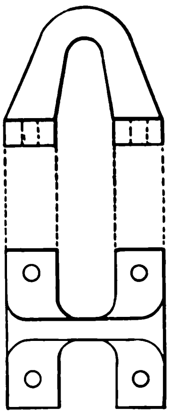

FIG. 51.

These last forms are objectionable for high pressures, on account of the uncertainty as to the strength of the tee and angle irons.

Locomotive Boiler. — The part of the head above the tubes is stayed in the same way as in a horizontal multitubular boiler.

The locomotive boiler, Plate V., has the upper part of the front head stayed by eight $1\frac{1}{8}$-inch diameter through stay bolts, together with four diagonal braces running to the second and third courses, and by two short diagonal braces. The sides of the fire box are stayed to the shell by screwed stay bolts headed over cold on the inside and outside. The holes for these have to be tapped with a special tap made long enough to reach through both plates.

The back end of the fire box between the lower tubes and the upper stay rivets in the water legs is generally stayed as shown in Fig. 1, Plate V. In this particular boiler, the second sheet being flattened at the sides, is stayed by three tee irons and cross rods as seen in Fig. 1.

The crown sheet being directly over the fire, and subjected to an intense heat, is the hardest part to stay properly. The ordinary methods used on the heads will not answer here, as overheating of the

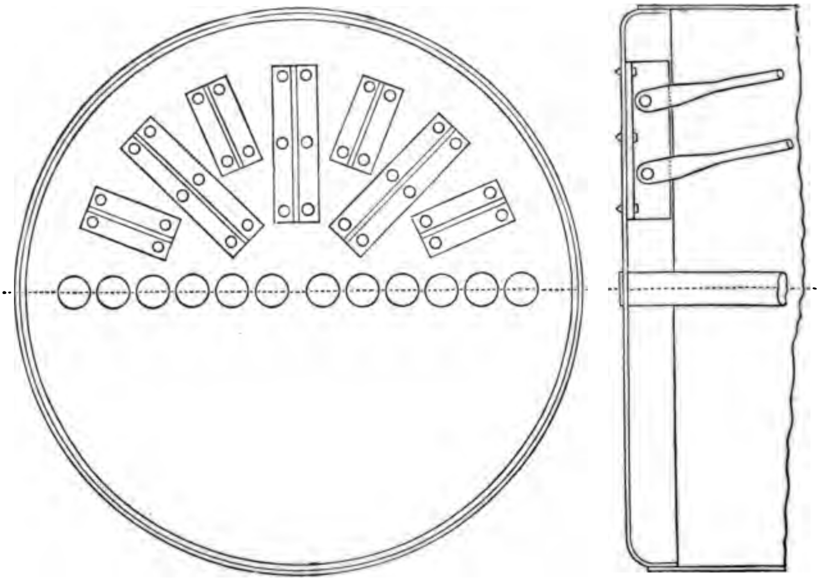

FIG. 52.

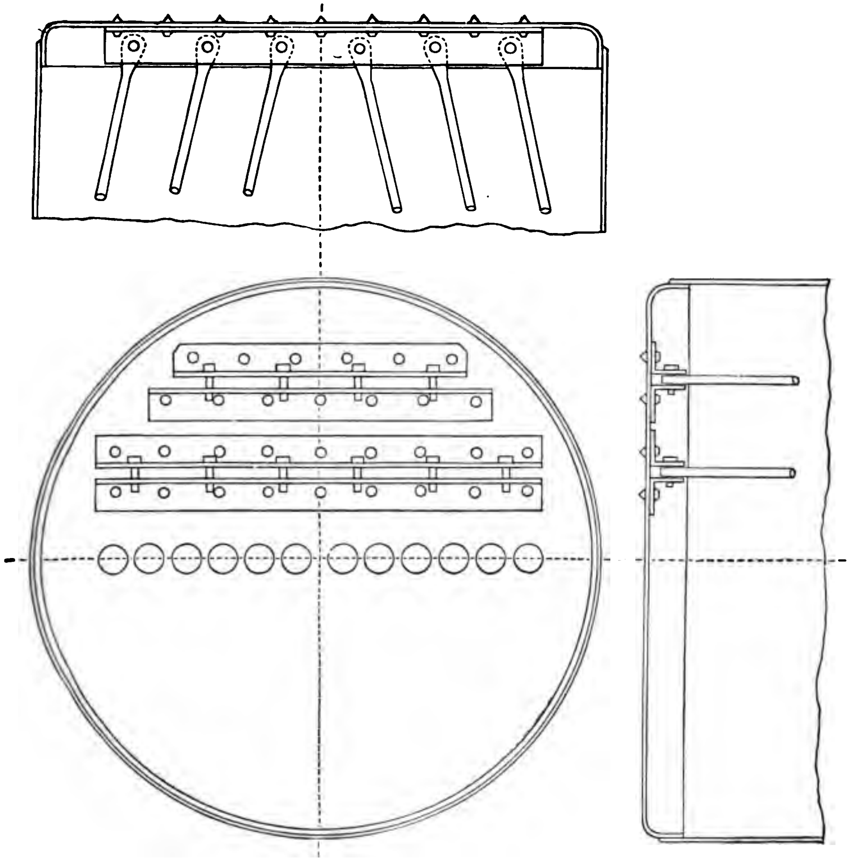

FIG. 53.

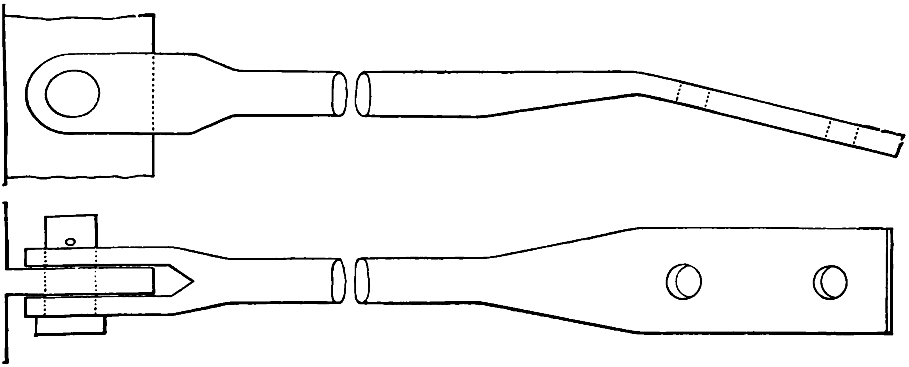

FIG. 54.

plates would occur under the channel bars or angles used because of the increased thickness of metal. On Plate V. Figs. 1 and 2 show the general arrangement and Figs. 5 and 6 details of crown-sheet staying. The crown sheet is supported by ⅞-inch bolts which carry this load to sets of crown bars supported by means of sling stays, and by forgings (Fig. 6) which rest on the edges of the plates forming the sides of the fire box. These crown bars are made of 4½″ × 1″ wrought iron.

The ⅞-inch bolts have copper washers under the heads, which serve to make a steam-tight joint with the plate. Under the nuts there are washers, as shown in Fig. 6, which keep the tops of two crown bars forming a set together. To prevent buckling the plate upward, from unduly tightening up the bolts in making a joint with the copper washer, ferrules of cast iron are put between the crown bars and the plate, as shown in Fig. 5. These ferrules are made quite small where they bear on the plate, so as to keep as little of the sheet as possible from direct contact with the water in the boiler. Each crown bar is hung in two places by sling stays attached either to the shell or to the sides of the dome. This reduces the span of the crown bars, which might otherwise become excessively heavy. The different sets of bars are all tied together at the top, and then the front set is tied to *the head.*

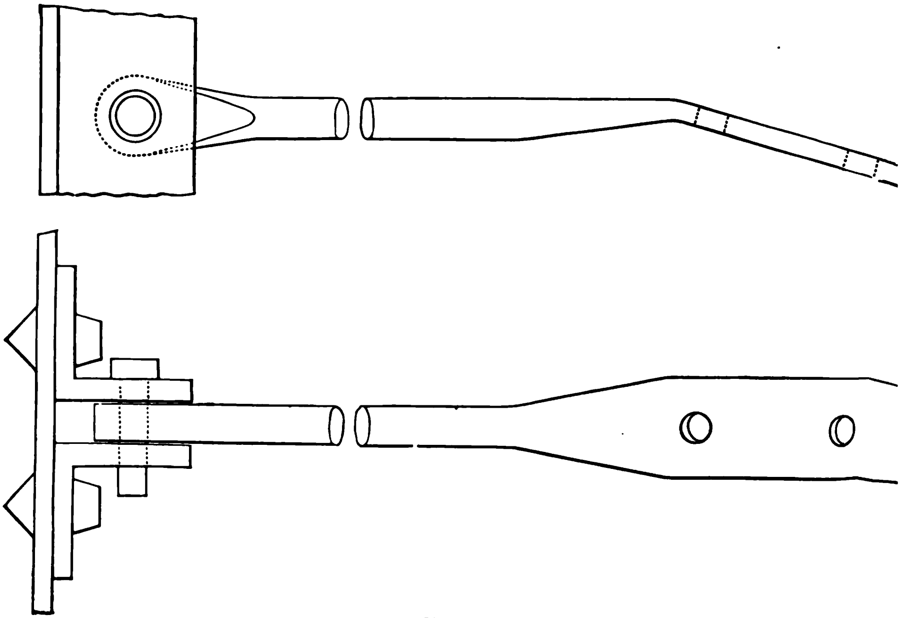

FIG. 55.

Crown bars are sometimes run along the length of the fire box or at right angles to those shown on Plate V.

Two other styles of supporting the crown sheet, now quite commonly used, are shown in Figs. 56 and 57. The stay rods run through the shell, where the joints are made by means of copper washers under the nuts. The joints in the crown sheet are made by conical enlargements, just under the heads of the stay rods, fitting into reamed conical holes in the plate. The tension on the stay rods, due to the pressure on the crown sheet, tends to make these joints fit closely.

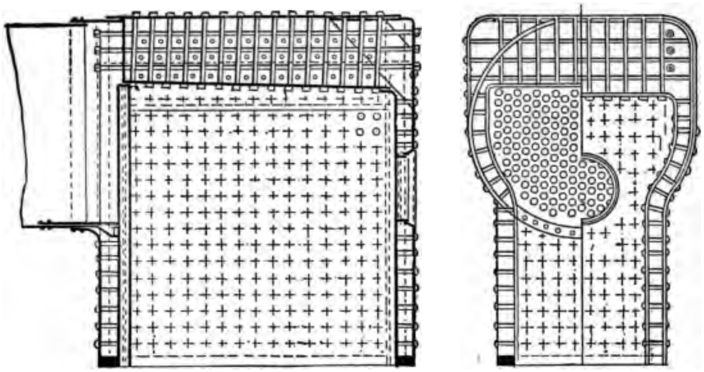

FIG. 56.

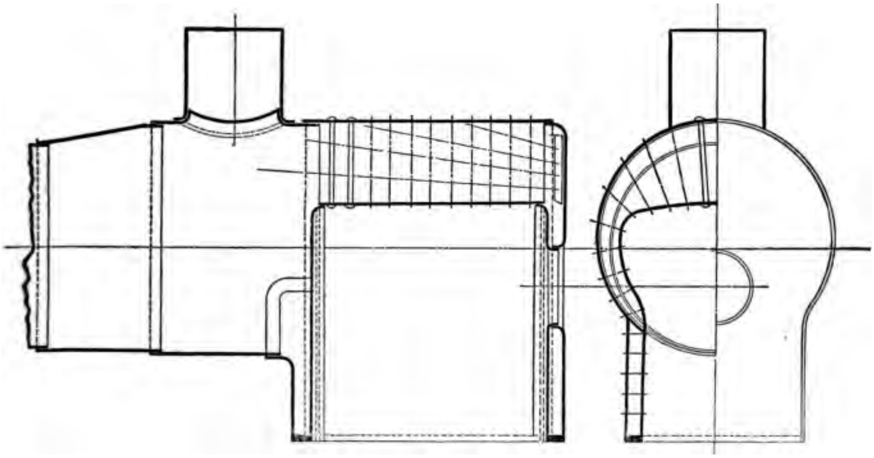

FIG. 57.

Marine Boiler.—Plate IV. shows a method of staying commonly used in Scotch boilers. As the heads are made of iron or steel, from $\frac{3}{4}''$ to $1''$ thick, the allowable distance between stay rivets is greater than in either locomotive or horizontal stationary boilers. That part of the heads above the tubes is stayed directly by 10 through rods $2''$ diameter. Thick washers, $8''$ diameter, hold the plate on the outside for some distance either side of the stay rods. The top part of the flat plate forming the front of the combustion chamber is held by the tubes. The lower part is tied to the front head by two rods of Burden's best iron, 2⅛ inches diameter, and five rods, 1¾ inches diameter.

The sides of the combustion chamber are tied to the shell by means of short stays placed as shown by the radial lines on the left side of the end view. Both front and back heads are stiffened between the tubes and the shell by tee irons bent to shape and riveted on. The flue tubes are subjected to external pressure, and of course have to be held in shape. This is done by corrugating the plates as shown in the longitudinal section. This also allows for expansion of the flues.

Gusset stays are sometimes used on the heads above the tubes. These are made of pieces of plate cut in the shape of a trapezoid, two sides being parallel and two at right angles. Each piece is fastened to two angle irons, one on the plate, and one on the head. These gusset stays, as seen in an end view, are placed on radial lines. Instead of corrugating flue tubes, they are often made plain and then strengthened by bands of wrought iron or angle irons, bent into a circle about 4″ larger than the flue, and joined to it by rivets running through ferrules (Fig. 58); the arrangement being similar to that of the ferrules used on crown sheets of locomotives.

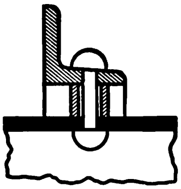

FIG. 58.

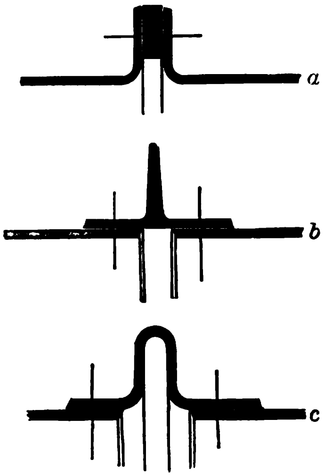

FIG. 59.

To allow for expansion, the different courses of plate in the flue tube are joined together through an expansion ring, which by its bending permits of some end motion. This ring should be kept from direct contact with the fire, and there should be as little thickening of the flue as possible at the joints. Figures 59*a* and 59*c* show two forms of joint, and Fig. 59*b* a joint which does not give any flexibility.

CHAPTER IX.

DETAILS.

Horizontal Multitubular Boilers. — Longitudinal seams are commonly double riveted, while roundabout seams, which have but one-half as much stress as the longitudinal, are single riveted.

Longitudinal seams are placed in the steam space with the joints in the different courses, alternating either side of the top of the boiler. The front course, which is directly over the fire, should be outside of the next course. This brings the roundabout lap pointing to the back end of the boiler, and the edge of the joint is thus protected from direct contact with the flame. The tubes occupy a little more than half of the head. They should be laid out in vertical rows, leaving a wider space at the centre, so as to allow the heated water to rise freely. Tubes are sometimes spaced diagonally; but, although more tubes may be put in by this method than by a vertical spacing, there is a poorer circulation of water. Entrance to the inside of the boiler is provided through a manhole in the middle course. It is important that the longer dimension of this hole should be along the circumference, and not along the length. The manhole cover is held in place by means of a bolt running through a cast-iron yoke. An inside manhole ring and cover is shown in Plate I.

An outside manhole ring and cover are shown in Fig. 50. A hand hole is made near the bottom of each head. The cover at the back head should be drawn up by means of a key rather than by a nut, as the thread is soon spoiled by the fire, rendering the removal of the nut difficult. Reinforce plates of boiler iron are riveted on for the feed and blow-off pipes. The blow-off pipe should have malleable fittings, and should have a plug cock placed outside of the setting. The feed pipe often leads under the water line, where it terminates in a perforated pipe running across the boiler.

A valve should be placed in the feed pipe close to the boiler, and outside of this valve a check valve, as shown in Fig. 86. Figs. 80, 83 show one form of stop valve, and check valve. Feed pipes are sometimes run through the front head. Two steam outlets are provided: one for a safety valve and one to draw from. The safety valve is generally placed on the front nozzle, as the steam is apt to be more moist here than at the back nozzle, where there is not such a violent boiling of the water. Connection is made to the steam nozzles by means of bolted

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flanges, into which the steam pipe screws. A pop safety valve is shown in Fig. 67 and a weighted lever valve in Fig. 66.

Boilers over 18 or 20 feet long usually have three brackets on a side. A very good form of bracket is shown in Plate I. The bracket is riveted on so that the bearing surface is a little above the centre of the boiler. Rollers are placed under all but the front set of brackets, to allow for the longitudinal expansion of the boiler.

A fusible plug, or, as it is often called, a safety plug, is screwed into the back head just above the top row of tubes. As long as this plug is under water, there is no danger of the fusible metal melting out; but if the level in the boiler drops below this point, the plug heats up, as steam will not carry away heat as rapidly as water, and finally melts out and allows steam to blow into the furnace, and thus give warning to the fireman.

A gauge-glass combination is placed in front of the setting, where it can be easily seen by the fireman. Figure 69 shows a common form.

There are gauge cocks on the side, which can be used instead of the glass to tell approximately the level of water in the boiler. The glass, although very strong, is quite brittle, and care has to be taken not to scratch it when putting it on, or in cleaning it.

Valves are placed at the top and bottom of the glass, by means of which an old glass may be replaced while steam is on the boiler. In case the glass should break, without any one being near to shut off the valves, the level of the water would be blown down through the lower cock to that level. Figure 70 shows a self-closing device often applied to the lower valve. The valves in the pipes leading from the combination to the boiler should be locked open, so that the combination shall at all times be open to both the steam space and water space. The steam gauge is sometimes attached to the top of the combination, but more often to the steam pipe or safety-valve nozzle. There should always be a siphon in the pipe under the gauge, so that only water goes into the tube in the gauge.

Locomotive Boilers. — The usual arrangement of plates is shown in Plate V. The smoke box is attached through a wrought-iron ring. Good methods of making the joint at the fire door are shown by Fig. 60.

Figure 61 shows the manner of making the joints at the steam dome. The manhole opening is in the top of the steam dome. Much smaller tubes are used than in horizontal multitubular boilers, the object being to get as much heating surface as possible; and for the same reason the tubes are often spaced diagonally.

Mud plugs, about 2″ diameter, are tapped into the plates near the bottom of the water legs. The plates which form the water legs are joined at the bottom to a rectangular ring of 4″ × 2½″ wrought iron drilled for rivets.

Steam is taken from the steam dome into a pipe running inside the boiler over the tubes, and passing through the head into the smoke box, where it is distributed through branch pipes to the cylinders on either side. A poppet valve, placed in the steam dome on the end of this pipe, controls the supply of steam to it.

This valve is connected through a bell-crank lever with a rod running over the crown bars out through the head into the cab.

The valve is nearly balanced. A slight excess of pressure on the larger disk tends to hold the valve shut.

The smoke-box end of a locomotive boiler is fastened firmly to the engine frame. The fire-box end, although tied to the frame, is left free to expand, as shown in Fig. 4, Plate V.

The exhaust from the cylinder being passed through the smoke stack, serves to keep up a strong current of air through the fire. When the engine is not running, a jet of live steam can be blown up the stack for the same purpose.

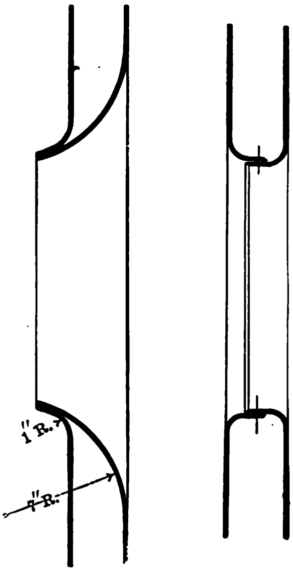
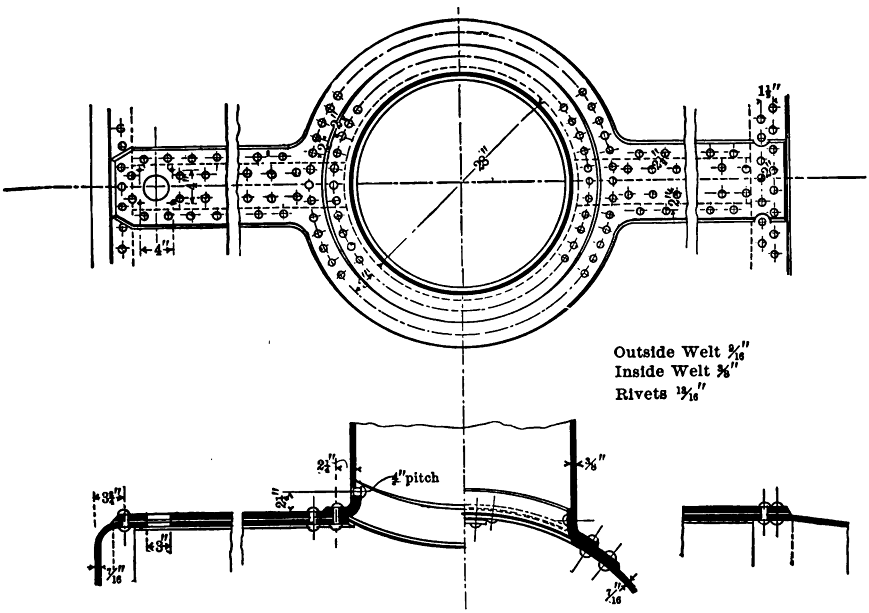


FIG. 60.



Outside Welt 9/16"
Inside Welt ⅜"
Rivets 13/16"

FIG. 61.

Marine Boilers. — In the Scotch boiler shown in Plate IV., each course is made up of three plates.

The heads are each made of two pieces, the joint in the front head coming just above the top row of tubes, and that in the back head at about the centre. The lower part of the front head is flanged outward at the flues, and inward at the edge.

Where the flue is nearest to the shell, the inward flange comes almost into the same line as the outward flange. Good material and great care are required to prevent heavy stock from cracking when flanging in this way.

A number of manholes and hand-hole openings give access to all parts of the flue tubes.

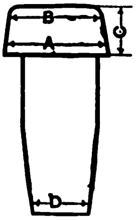


FIG. 62.

Steam is drawn from the boiler through a perforated dry pipe running in the steam space the whole length of the boiler, and passing out through the front head.

An uptake made of thin plate is bolted to angle irons riveted to the front head. This uptake is fitted around the flue tubes so as to include all the tubes.

Rivets. — Figure 62 illustrates a pan-tailed rivet such as is used in machine riveting.

In Fig. 63, *A* shows a conical head such as is made by hand riveting.

B shows a cup-headed or snap-headed rivet, formed by hammering down the point, and finishing with a cup-shaped die.

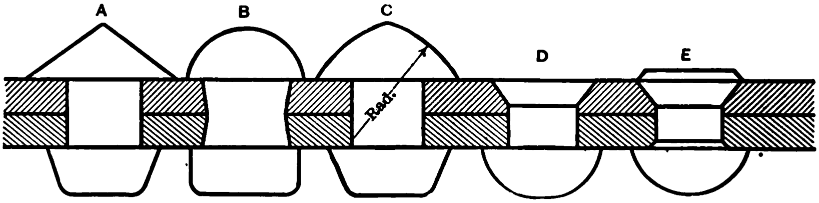


FIG. 63.

B also shows how punched plates should be put together.

C has a machine-riveted head. The inner head is sometimes made the same as the outer.

D and *E* show countersunk heads.

CHAPTER X.

CALCULATION OF STRENGTH OF BOILERS.

THE calculation of the stresses coming on the different parts and members of a boiler, are made by the application of the principles of applied mechanics; the method of application will be given here, together with a few special problems met in designing boilers.

Thin, Hollow Cylinders. — If p is the pressure of the steam (or other fluid) in a thin, hollow cylinder having the thickness t and the radius r, then the stress per square inch along an element of the cylinder will be[1]

$$T = \frac{pr}{t};$$

for safety, the stress T should not exceed the working stress f, so that

$$f = \frac{pr}{t}, \qquad t = \frac{pr}{f}.$$

The stress on the section of the cylinder made by a plane at right angles to its axis may be found as follows: —

Suppose the end of the cylinder is a thick, flat plate, then the pressure of the fluid in the cylinder on the end is (using the same notation as before)

$$\pi r^2 \cdot p;$$

this total stress is resisted by an annular ring of metal having the approximate area

$$2\pi rt,$$

so that the stress per square inch is

$$T = \frac{\pi r^2 p}{2\pi rt} = \frac{p}{2rt},$$

and in order that the working stress shall not be exceeded, we should have

$$f = \frac{p}{2rt}, \qquad t = \frac{p}{2rf}.$$

---

[1] Applied Mechanics, Lanza, page 252.

But the force required to hold on the end of the boiler does not depend on the form of the end; so the above equations apply to any thin, hollow cylinder.

Since the stress along an element is twice the stress on a section perpendicular to the axis, it would appear to be unnecessary to make the latter calculation for a homogeneous cylinder of uniform thickness. But boilers are made by riveting plates with longitudinal and ring seams, and the real strength of the cylinder will depend on the efficiency of the riveted joints used; consequently both calculations should be made. The ring seam has usually a less efficient joint than the longitudinal seam, but the efficiency is easily made more than half as much, so that a new boiler will always be stronger on the ring seam.

Example. — A thin, hollow cylinder, 5 feet in diameter and $\frac{1}{2}$ an inch thick, under the pressure of 100 pounds to the square inch, will be subjected to a stress along an element of

$$5 \times 12 \times 100 \div \tfrac{1}{2} = 12{,}000 \text{ pounds}$$

per square inch; if the cylinder were made of one plate of steel without a seam, this stress would be about $\frac{1}{5}$ of ultimate strength.

Thin, Hollow Sphere. — The stress on a diametrical section of a thin, hollow sphere is the same as the stress on a section of a thin, hollow cylinder of the same diameter and thickness, made by a plane perpendicular to the axis.

Formerly the ends of plain cylindrical boilers were made hemispherical, but such ends are difficult to make and are needlessly strong, if of the same thickness as the cylindrical shell. It is now the practice to curve such ends to a less radius than that of the cylindrical shell. If the radius of the head is equal to the diameter of the shell, as shown in Fig. 64, then, with the same thickness of plate, the stress will be the same per square inch, provided there are no seams in head or shell. The heads usually do not have a seam, and the shells always have a seam; the margin of strength in the head, when the same thickness of plate is used, under this condition may be offset against the possible injury done to the head in shaping it.

The construction known as a *bumped-up head*, has the edge flanged into a cylindrical form to make a joint with the shell, and to avoid the awkward stress that would be thrown onto the cylindrical shell if the true cylindrical and spherical surfaces were allowed to intersect.

If it is inconvenient to curve the head to a radius as small as the diameter of the cylinder, then a thicker plate may be used, with a longer radius.

Riveted Joints. — An adequate knowledge of the properties of riveted joints for a given purpose can be obtained only by direct tests *on full-sized* pieces. A series of such tests made for the Bureau of

Steam Engineering,[1] showed in all cases a failure by shearing the rivet or by tearing the plate between the rivets, and such a result is to be expected for all joints used in boiler making, where the pitch of the rivets is regulated by the necessity of making a tight joint by calking. For joints, like that shown by Fig. 65, which have a wider cover plate on the steam side than on the outside, the outside row of rivets *may* give an excessive compression on the bearing surface in front of the rivets, and in such case it is well to make a calculation for the compression by the common method.[2] Experiments on riveted joints for boiler work show that the apparent compression on the plate in front of the rivets, as calculated by the common method, is always greater than the known compressive strength of the plate, when the joint failed by compression, and that in some cases the apparent compression is twice the compressive strength of the plate. It will be proper to assume, in such calculations, the factor for compression in the common formula to be not more than 90,000 pounds.

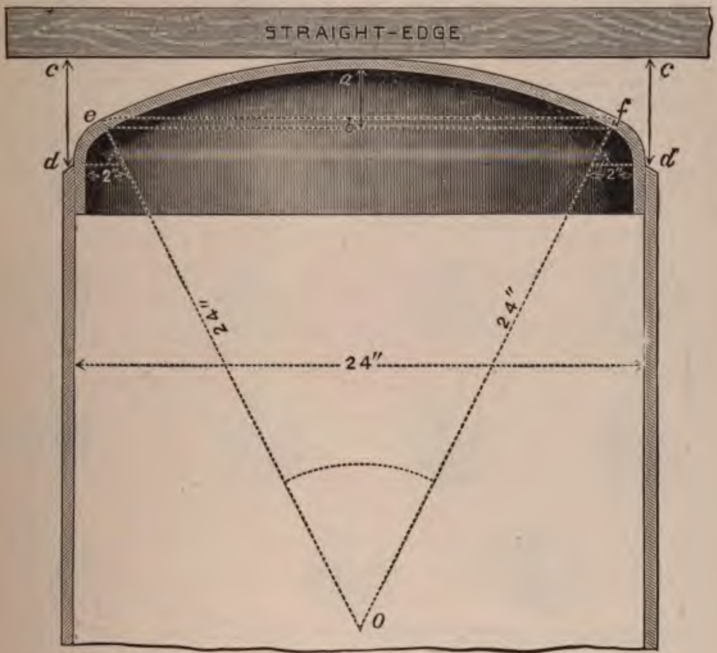


FIG. 64.

The lap of a riveted joint for boiler work can always be made enough to avoid shearing or breaking of the plate in front of the rivet, and it always is enough in good practice. Calculations for failure by these methods may be advisable for a joint of unusual proportions, using the common method already referred to.

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[1] Applied Mechanics, Lanza, page 625.

[2] *Ibid.*, page 258.

The efficiency of a single-riveted lap joint calculated for shear of the rivets and tearing of the plate between rivets will commonly be 56 per cent, and of a double-riveted lap joint 70 per cent; special joints, similar to that shown by Fig. 65, but with two rows of widely spaced rivets outside of the narrow cover plate, have shown by experiment an efficiency of 85 per cent.

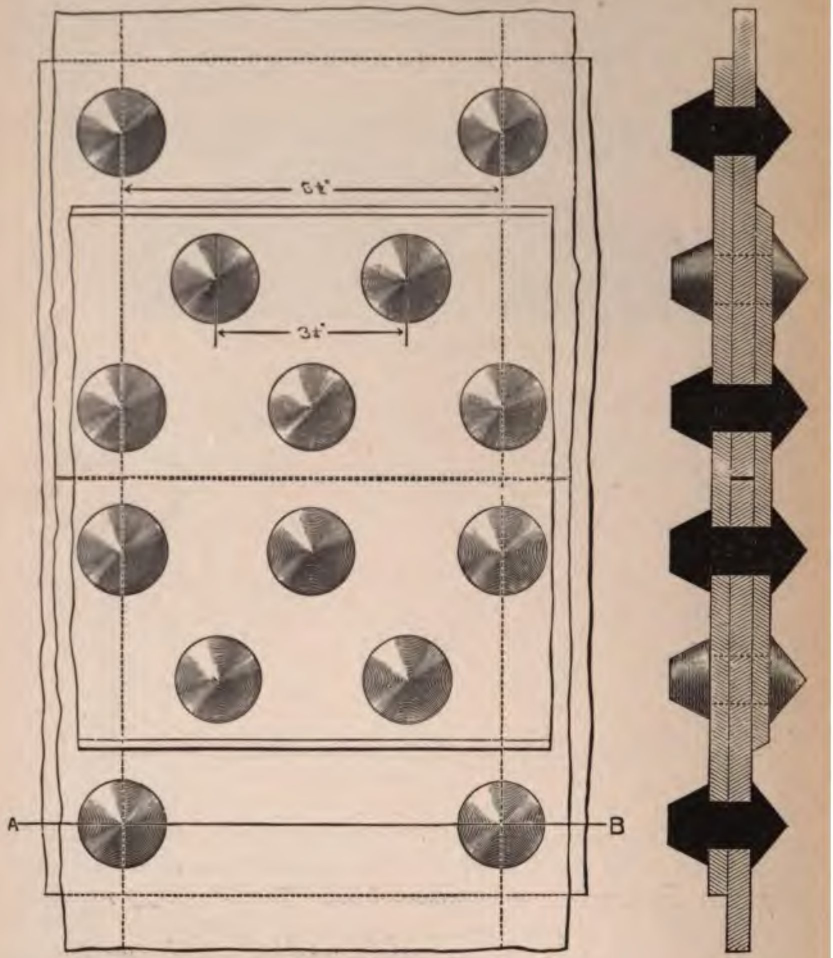

Fig. 65.

Tables of common proportions for riveted joints will be found on pages 529 to 535 of Lanza's *Applied Mechanics*.

Flat Plates. — The stress on flat plates used sometimes in boiler work can be calculated only by the aid of the theory of elasticity; the student is referred for the discussion and the resulting equations to the tenth chapter of Lanza's *Applied Mechanics.* The following cases are liable to occur: —

(1) *Round, flat plate supported all round.* — For example, flat cast-iron heads for plain cylindrical boilers or drums. The objections to the use of cast iron for boiler making have almost entirely excluded this form of construction from good practice. There does not seem to be the same objection to the use of flat plates of iron or steel if special considerations warrant the use of plates of the necessary thickness. Although such plates are riveted to cylindrical shells having some rigidity, they cannot be considered to be fixed at the edges, nor can the effect of the rigidity of the shell be allowed for.

(2) *Stayed flat plates.* — This case is treated in the theory of elasticity, under the head of "indefinite plates which are firmly held at a system of points dividing them into rectangular panels." The only condition that leads to a definite solution is that the panels shall be squares; this is the common condition when numerous stay bolts are used. Should it be convenient to space stay bolts so as to give rectangular panels with unequal sides, then the longer dimension may be used in the equation for square panels; the stress thus calculated will be greater than the actual stress by an unknown amount, but the error is on the side of safety. There does not appear to be any way of calculating the stress on a plate stayed at irregular intervals.

(3) *Sheets of a locomotive fire box.* — The side sheets of a locomotive fire box very commonly carry part of the load, resulting from the pressure of the steam, on the top of the fire box; this gives rise to a compression which must be compounded with the stresses arising from the staying of the sheet.

Stays. — The stays which support flat plates are either round rods, as shown in Plates I. and IV., or are flat plates, shown by Fig. 3, page 5, as applied to a Lancashire boiler.

When straight stays, like those passing through the steam space of the Scotch boiler shown by Plate IV., are evenly spaced, each stay carries the pressure of the steam on the rectangle at the centre of which it is placed. In the case mentioned, the stays are spaced 14 inches from centre to centre horizontally, and 13 inches vertically, so that with a pressure of 160 pounds to the square inch, the pull on the stay is

$$14 \times 13 \times 160 = 29{,}120 \text{ pounds.}$$

The diameter of the rod is 2 inches, and the area is 3.14 square inches; so that the stress per square inch is

$$29{,}120 \div 3.14 = 9300$$

pounds per square inch.

If stays are run diagonally to the side of the shell, as in Fig. 12, page 10, the pull on the stay is evidently greater than it would be if the stay were perpendicular to the surface it supports. The allowance for the angle at which the stay is placed may usually be made by a simple proportion. For example, let a diagonal stay, for the flat end of a cylindrical boiler, be attached at a point 2 feet from the shell, and let it meet the shell 6 feet from the head. Then the pull will be greater than on a through stay, in the proportion

$$\frac{(2^2 + 6^2)^{\frac{1}{2}}}{6} = 1.054.$$

If stays support a girder, as, for example, the girder riveted to the head of the cylindrical boiler shown by Plate I., the pull on the stays will depend on the distribution of the load on the girder. This will be discussed in connection with the calculation for such a girder.

Stay Bolts are usually arranged in rows, so that each supports the pressure of the steam on a square or rectangle, and the calculation is like that for through stays on page 107.

For example, the stays for the locomotive fire box on Plate V. are spaced 4 inches on centres and are $\frac{7}{8}$ of an inch in diameter. The area of cross section may be taken to be 0.6 of a square inch, so that with 170 pounds steam pressure, the stress per square inch of the stay appears to be

$$4 \times 4 \times 170 \div 0.6 = 4500 \text{ pounds.}$$

In addition to the direct stress on a stay bolt, there is liable to be an awkward cross-breaking stress, due to unequal expansion of the inner and outer shells of the fire box; consequently the apparent working stress is always kept low. The screw thread between the plates should be turned off to avoid the bad effect of the sharp V-shaped groove along the bottom of the screw thread; when this is not done, there is additional reason for restricting the working stress. If the outside diameter of the stay over the threads is known only, and if that diameter is used in calculating the area of the stay, the apparent stress will be about 0.8 of the real stress. The real diameter should always be known and used.

Gusset Stays are open to the defect that the distribution of stress on the plate forming the stay is uneven and uncertain. It is customary to calculate them on the assumption that the resultant stress acts along a medial line, and is evenly distributed over the smallest section at right angles to this line. A low apparent working stress should be used.

Girders. — When it is not convenient to stay a flat surface to the opposite shell of the boiler, it may be stayed to girders that are properly supported at the ends. As example, we may refer to the flat plates above the fire tubes in the cylindrical tubular boiler on Plate I.; to the crown sheet of the fire box of the locomotive boiler on Plate V.; and to the top of the combustion chamber of the marine boiler on Plate IV.

The last case mentioned is the most definite and simple, and will be chosen for calculation. Ordinarily it would be proper to assume the supporting forces to act at the middle of the feet at the ends of the girder, but in this case the girders are supported on the flanged ends of the tube sheet and the back sheet of the fire box, and it is possible that the plates may yield in such a way as to throw the supporting forces to the extreme ends of the girder. The girders are spaced 7 inches apart, as shown by the cross section Plate IV., and the stay bolts are spread $6\frac{1}{4}$ inches apart, as shown by the longitudinal section. The load on each stay bolt is consequently at 160 pounds steam pressure,

$$7 \times 6\tfrac{1}{4} \times 160 = 7000 \text{ pounds},$$

and the total load on the girder is 21,000 pounds. The supporting forces are each 10,500 pounds. The bending moment at the middle of the girder is

$$10{,}500 \times \tfrac{25}{2} - 7000 \times 6\tfrac{1}{4} = 131{,}250.$$

Each plate of the pair forming a girder is $\frac{5}{8}$ of an inch thick and 7 inches deep. If I is the moment of inertia of the section of both plates, and if y is the distance of the most strained fibre from the neutral axis, then the greatest fibre stress is

$$f = \frac{My}{I} = \frac{131{,}250 \times 3\frac{1}{2}}{\frac{1}{12} \times 2 \times \frac{5}{8} \times 7^3} = 12{,}857$$

pounds per square inch.

The crown sheet of the fire box on Plate V. is also supported by a series of double girders, but these are slung from the shell as well as supported at the ends. Even with careful workmanship there is considerable uncertainty about the distribution of the load on the supports. If the supports are assumed to be rigid, the calculation of the supporting forces and of the stresses on the girder may be made by the general theory of continuous girders.

The girders shown at the end plates of the cylindrical boiler on Plate I. are in somewhat the same case, since the ends beyond the through stays receive some support from the adjacent cylindrical shell.

On page 316 of Lanza's *Applied Mechanics*, it is shown that the point of inflection of a beam fixed at the ends and uniformly loaded is

$$\frac{1}{2} - \frac{1}{2\sqrt{3}} = 0.211$$

of the span from one end. If we have a girder supported at uniform intervals, and if the ends project 0.21 of the uniform interval beyond the end supports, then with a uniformly distributed load, the supporting forces will be equal and the girders may be calculated as a beam fixed at the ends. If the projecting ends have a different length, then the general theory for a continuous girder must be used.

Flues. — The tendency of internal pressure in a thin, hollow cylinder is to give it a true cylindrical shape; consequently, with fair workmanship, the formulæ for thin, hollow cylinders may be applied to the calculation of boiler shells subjected to internal pressure. But the tendency of external pressure is to exaggerate any imperfection of shape, and cylindrical flues fail by collapsing. The pressure at which a flue will collapse can be found by direct experiment only. The following table, given by Hutton,[1] gives the results of some experiments to show the collapsing pressure: —

EXPERIMENTS ON THE COLLAPSING PRESSURE OF BOILER FLUES.

Where or By Whom Made.	Dimensions.			Collapsing Pressure in Pounds per Square Inch.		
	External Diameter in inches. 2	Length in inches. 3	Thickness in 32ds of an inch. 4	Found by experiment. 5	Calculated by Fairbairn's rule. 6	Calculated by Hutton's rule. 7
By Fairbairn	7.87	276	5	110	109	114
By Fairbairn	33.5	360	11	99	81	113
By Fairbairn	42	420	12	97	78	100
By Fairbairn	42	300	12	127	108	119
Engineering Dept., U. S. N.	54	36	8	128	311	120
At Greenock	38	86	16	450	740	436
By Knight	36	24	8	235	700	218
By Knight	36	24	12	468	1568	490
By Knight	36	48	12	390	784	350
By J. Howden & Co., Glasgow	43	23	17	840	2758	842

As a result of his experiments, Fairbairn prepared the following rule for calculating the collapsing pressure of flues: —

$$p = \frac{806{,}300 \times t^{2.19}}{l \times d};$$

in which l is the length of the flue in feet, d is the diameter in inches, and t is the thickness in inches. Sometimes the exponent of t is made

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[1] Steam Boiler Construction, page 180.

2 instead of 2.19, for convenience in calculation. Since all of his flues were long and collapsed at moderate pressure, it is evident that the equation should not be applied to shorter flues which collapse at much higher pressure. The comparison of the experimental results on shorter flues (column 5 of the table), with the values calculated for these cases by Fairbairn's rules (column 6), makes this evident.

Hutton proposes the following rule: —

$$p = \frac{Ct^2}{d\sqrt{l}};$$

in which l is the length in inches, d is the external diameter in inches, and t is the thickness in *thirty-seconds* of an inch, C is a constant which is 600 for wrought iron and 660 for mild steel. The seventh column of the table was calculated by this rule; the later experiments are represented very nearly by this rule; in one case only is the calculated result in excess of the experimental result. Fairbairn's experiments, however, are not well represented, as the rule gives higher results; in two cases much higher. Mr. Michael Longridge, as the result of an investigation of numerous boiler flues, most of which have endured service for years, but some of which failed, gives a rule in the same form but with a constant 540 instead of 600.

For oval flues it is recommended that the rule be applied, using for d twice the maximum radius of curvature.

Fire Tubes. — The thickness usually given to fire tubes to ensure sound welding and to provide for expanding into the tube sheets, is in excess of that required to prevent collapsing. There appears, however, to be no experiments to show the actual collapsing pressure for such tubes.

The joint made by expanding the tubes into the tube sheets of locomotive and cylindrical tubular boilers, has been found both by experiment and practice to be strong enough to secure the tube sheet without additional staying. It is, however, the custom to make part of the fire tubes of marine drum boilers thick enough to take a shallow nut outside of the tube plate; without such stay tubes there is liable to be leakage at the ends of the tubes.

Factor of Safety. — The ratio of the working pressure of a boiler to the pressure at which the boiler or any part of a boiler may be expected to fail quickly, is called the factor of safety for the boiler or for that part of the boiler.

It is commonly recommended by writers that a factor of safety of six shall be used for boilers; probably such a factor would be economical for a boiler that is expected to work continuously for many years, as it allows a margin for deterioration. If the stresses coming on the parts of a boiler can be determined, a general factor of five will give sufficient security. If the boiler is carefully watched, a factor of four may be used; many boilers are worked with this factor. Of course parts like stays, which are subject to more corrosion than the shell, should have a larger factor. Moreover, such parts may commonly be given ample strength without adding much to the cost or weight of the boiler. Attention has been called to this in connection with the discussion of the strength of the various parts of the boiler.

Hydraulic Tests of Boilers. — It is customary to subject new boilers to a water pressure considerably in excess of the working pressure, to discover any leaks at riveted joints at the tube sheets or elsewhere; should there be any gross defect of design or workmanship, it will be developed by this hydraulic test. Old boilers after repairs are subjected to a hydraulic test for the same purpose, but the pressure is not carried so high as for new boilers.

The pressure applied during a hydraulic test is seldom more than once and a half the working pressure, and as most boilers have an actual factor of safety of not more than five, and frequently of four, it is apparent that the recommendation of some authors, that the test pressure should be twice the working pressure, cannot ordinarily be followed without danger of injuring the boiler. With a factor of safety of six, there should be no danger of injuring the boiler by applying a hydraulic pressure twice the working pressure.

It should be borne in mind that some of the worst stresses that come on the different parts of the boilers are due to unequal expansion and contraction, and that such stresses are not set up during a hydraulic test. Finally, the fact that a boiler has successfully withstood a hydraulic test is not a conclusive proof that it is safe; too many unfortunate explosions of boilers, more frequently old boilers, after a hydraulic test have shown this.

The safety of a boiler is to be ensured by careful and correct design, honest and thorough workmanship, and intelligent care in service. Forms and methods of design and construction that do not admit of ready calculation should be avoided; in no case should the ordinary hydraulic test be relied upon to guarantee the strength of parts that cannot be calculated with a fair degree of certainty. If such forms are used in any case, they ought to be tested separately to a pressure of two or three times the working pressure, and some examples of each form and size ought to be tested to destruction.

The boiler undergoing a hydraulic test should be carefully inspected, and any notable change of shape or leakage should be investigated to discover the cause. Frequently small leaks that are developed during a test are stopped at once by caulking or otherwise, but it is preferable to mark the place of the leak and caulk after the pressure is removed. This, of course, requires another test to find out if the caulking is successful.

The pressure is usually applied by filling the boiler entirely full of water and then pumping in enough water, by hand or by power, to supply the leaks and develop the pressure required. If the pumping is done by hand, it is desirable to carefully remove all air from the boiler to avoid the labor of compressing air up to the test pressure. If the pumping is done by power, the saving of work is of less consequence, and a little air remaining in the boiler will act as a cushion and lessen the shocks due to the strokes of the pump.

New boilers are tested on the boiler-shop floor; old boilers are commonly tested in their settings, and in such case the inspection during a test is less convenient and efficient.

It is sometimes recommended that hot water shall be used for testing a boiler; but there seems to be no advantage in so doing, as it is unequal expansion, and not merely rise of temperature, that sets up the unknown stresses that are so destructive to the boiler. Of course the use of hot water makes an efficient inspection, during the test, difficult if not impossible.

When there is no other way of applying the hydraulic test to a boiler in its setting, the boiler may be quite filled with water, and then a light fire may be started in the furnace. The expansion of the water will develop the required pressure at a much less temperature than that of steam at the same pressure, and with less danger should the boiler fail. This method cannot be recommended for general use; and in case it is followed, care must be taken not to exceed the desired pressure.

Hydraulic Test to Destruction. — In 1888 a boiler shell, made to represent a part of the shell of a gun-boat boiler, was tested by hydraulic pressure at the Greenock Foundry[1] with the intention of bursting it. The shell was 11 feet long and 7 feet $8\frac{3}{16}$ inches mean diameter. It was made of three sections of $\frac{19}{32}$ plate, triple riveted with butt joints and double cover plates at the longitudinal joints, and lapped and double riveted at the ring seams. The rivets were staggered for both longitudinal and ring seams. The end plates were $\frac{20}{32}$ thick and stayed with through stays and washers, spaced 14 inches on centres. The stays were $1\frac{1}{8}$ inches in diameter; the screws at the ends of the stays were $2\frac{1}{4}$ inches in diameter. Finally, it may be said that the shell was designed to fulfil the admiralty specifications for a working pressure of 145 pounds per square inch. The workmanship was of the same degree of excellence usual for boiler work at that establishment.

First Test. — The shell was first subjected to the working pressure of 145 pounds, and showed a slight alteration of form due to the tendency of internal pressure to give it a true cylindrical form. The pressure was then raised to the admiralty test pressure of 235 pounds, and then to 300 pounds without developing leaks. There were some minor changes of form due to the increase of pressure. The pressure was then removed and the shell returned to its original dimensions.

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[1] Trans. Inst. Naval Arch., Vol. xxx., page 285.

Pressure was then raised to 330 pounds, when there was a slight leak at the manhole door. At 450 pounds pressure, the leak at the manhole door exceeded the capacity of the pumps. There was also a slight leak at the corners of two butts. The manhole was then strengthened — no other repairs were made.

Second Test. — Pressure was raised to 350 pounds and developed a small leak at the manhole. There were slight leaks at the butt straps, which were caulked at the end of the test. The manhole, however, leaked so that the test was stopped.

Third Test. — After additional bolts were put into the manhole cover, the pressure was raised to 350 pounds without leakage. At 360 pounds, the manhole began to leak, and at 580 pounds, the test was stopped on that account. The butt straps opened visibly at the caulking and leaked more than before.

Fourth Test. — The butt joints were again caulked and additional pumps were employed. The shell was again tight at 350 pounds and the pressure was carried to 620 pounds, at which there was a good deal of leakage at the butt straps. Only one or two rivets showed signs of leakage; there appeared to be no difference between the hand and machine riveting in this respect. At the pressure of 620 pounds, the entire capacity of the pumps was required to supply the leakage.

The distortion of the shell was very marked at the higher pressures, and increased with the pressure; thus, the ends bulged an inch at 520 pounds, about $1\frac{1}{2}$ inches at 580 pounds, and nearly two inches at 620 pounds. The sides bulged more irregularly, but to the extent of nearly an inch at 620 pounds. The stays drew down uniformly $\frac{1}{64}$ of an inch at 520 pounds, $\frac{2}{64}$ at 580 pounds, and $\frac{4}{64}$ at 620 pounds. They increased in length $2\frac{1}{32}$ inches at 520 pounds, $3\frac{1}{8}$ inches at 580 pounds, and $3\frac{5}{8}$ inches at 620 pounds; this accounts for the bulging of the end plates.

The mean tensional strength of the plates from which the shell and butt straps were made may be taken at 61,500 pounds. At 620 pounds the tension on the plates between the rivet holes was 57,504 pounds, or $93\frac{1}{2}$ per cent of the strength of the solid plate, and there was no serious disturbance of the structure. The ring seams increased in diameter about $\frac{5}{8}$ of an inch, and the shell bulged out between them.

The various portions of the boiler acted in harmony and showed no special weakness at any point. The butt joints had the rivets spaced $5\frac{3}{4}$ inches on centres to give a percentage of 83.7 per cent of the plate, and this may have caused the leakage found there. The riveting appeared to be reliable at the extreme pressure reached. This test *seems to show* that a boiler will give signs of weakness long before it will fail. Such signs of weakness should be carefully investigated; if there is any local weakness or deterioration, repairs or alterations may be made; if there are evidences of general deterioration, the working pressure must be reduced, or better, the boiler may be replaced by a new one.

Boiler Explosions. — The great destruction of life and property that is liable to be caused by a violent boiler explosion makes it imperative that the causes should be carefully investigated, to the end that explosions may be prevented.

An idea of the energy stored in the steam and water in a boiler may be obtained by a calculation of the intrinsic energy in the steam and water in the boiler. Thus, each pound of steam at the pressure p_1 will have an intrinsic energy reckoned from freezing point

$$778(\rho_1 + q_1)$$

foot pounds, in which ρ_1 is the heat equivalent of the internal work, and q_1 is the heat of the liquid corresponding to the pressure p_1. Each pound of water, at the temperature t_1 corresponding to the pressure p_1, will have the intrinsic energy: —

$$778\, q_1.$$

For example, the cylindrical tubular boiler on Plate I. will contain about 6600 pounds of water and 22 pounds of steam at 150 pounds pressure by the gauge. The intrinsic energy in the steam will be

$$22 \times 778(338 + 772) = 19{,}000{,}000 \text{ foot pounds.}$$

The intrinsic energy in the hot water will be

$$6600 \times 778 \times 338 = 1{,}736{,}000{,}000 \text{ foot pounds.}$$

The first point to be noticed is that there is much more energy in the water than in the steam, and the second is, that even a fraction of this energy is sufficient to account for the destruction caused by a boiler explosion.

The circumstances of the boiler explosion will determine how much of this energy is developed and how it is applied. Even in a particular case it is seldom possible to make proper estimates, nor does there appear to be any advantage from so doing. The following calculation may, however, be interesting. The total weight of water and steam in the boiler chosen is

$$6600 + 22 = 6622 \text{ pounds,}$$

and the value of x_1 is

$$22 \div 6622 = 0.00329.$$

An adiabatic expansion to the pressure of the atmosphere p_2 will give for x_2

$$x_2 = \frac{T_2}{r_2}\left(\frac{x_1 r_1}{T_1} + \theta_1 - \theta_2\right) = \frac{673.7}{965.1}\left(\frac{0.00329 \times 855.6}{826.6} + 0.5230 - 0.3143\right)$$
$$= 0.148.$$

The intrinsic energy in the water and steam after such an adiabatic expansion will be

$$6622 \times 778(x_2 \rho_2 + q_2) = 6622 \times 778(0.148 \times 892.6 + 181.8)$$
$$= 1{,}617{,}000{,}000.$$

The change of intrinsic energy, which would be transformed into external work if the expansion took place in a non-conducting cylinder, is therefore

$$1{,}736{,}000{,}000 + 19{,}000{,}000 - 1{,}617{,}000{,}000 = 138{,}000{,}000 \text{ foot pounds.}$$

While various causes may lead to boiler explosion, it is unfortunately true that by far the greater part of violent explosions are due to the fact that the boiler is too weak to endure service at the regular working pressure. A new boiler may be weak through defective design or workmanship; there can be no excuse for the explosion of a new boiler from weakness, and such explosions in good practice are rare. An old boiler is liable to become weak through local or general corrosion or other deterioration; this amounts to saying that a boiler will eventually wear out.

Hutton gives for the life of steam boilers that have been kept in good repair, the following record: —

Egg-ended boilers	40 years
Lancashire boilers	33 "
Locomotive boilers	24 "
Multitubular boilers, various	22 "
Portable boilers, locomotive type	20 "
Return-tube boilers	18 "
Water-tube boilers	15 "

He further says that the average life of a locomotive boiler is ten years, but that the fire box needs renewal after from three to six years. Also that marine boilers have in some cases worn out in twelve years, and in some have lasted longer.

If boilers are subjected to careless or ignorant abuse, they may be used up in a fraction of their proper time of service, especially if cheaply built. This will account for the numerous explosions of sawmill boilers and agricultural boilers.

It has been pointed out that leakage is frequently a sign of weakness; a perversion of this idea leads to the assumption that a boiler is safe as long as it can be kept from leaking. Too many boiler explosions have this history: The boiler, after long and satisfactory service, began to leak; a cheap man was employed to repair the boiler, the repairs consisting mainly of excessive caulking to stop the leaks; soon after the repairs, perhaps the first time the boiler was fired up, it exploded violently. A fit conclusion of the history is to ascribe the explosion to some obscure cause or to carelessness of the attendant, if he was killed by the explosion.

Serious injury may be caused by overheating any part of the heating surface, due to low water, to defective circulation, or to deposits of non-conducting substance on the plates or tubes. The overheated member, or plates, of the boiler may burst or collapse, and such failure may lead to an explosion of the boiler, but frequently the escape of steam and water will check the fire and relieve the pressure on the boiler. Local failures are dangerous to the boiler attendants, especially in a confined fire room, as on shipboard. Unless there is direct evidence of overheating, either from known circumstances before the explosion or from signs on the boiler after explosion, the cause of the failure should be sought elsewhere.

If a boiler shows signs of low water or overheating, the fire should be checked and the steam pressure reduced by any effectual means. The most ready way of checking the fire is to close the ash-pit doors and to throw ashes, or even fuel, onto the fire; if fuel is thrown on, the fire should be drawn or extinguished before the fuel becomes well ignited. The attempt to draw the fire without first deadening, as described, is liable to give a fierce combustion for a short time. If the furnace has a dumping grate, the fire may be thrown into the ash pit immediately. It is best to have the damper open, so that escaping steam may pass up the chimney instead of out into the fire room.

The steam pressure may be relieved by opening the safety valve and turning on the feed water. Both of these methods have been objected to, but on unsufficient grounds. If the steam in a boiler has been raised to nearly the bursting pressure, then a sudden opening of a large safety valve may determine the explosion, as has been shown by experiment. This may cause some hesitation about opening the safety valve if the pressure on a boiler is excessive, but not if there is danger of failure from overheating. In case the safety valve is not open and can be opened, and if the boiler pressure is above the working pressure, it should be opened, for if the boiler is so near explosion that the opening of the safety valve will determine the explosion, there is usually no time even to run away.

If the feed water is introduced in such a way that a large influx may give rise to local strains from contraction, then it may be dangerous to turn on the feed water to a boiler that is on the point of explosion. But this can hardly apply to low water or overheating, for then the danger is local, and if the feed water is directed immediately onto the overheated plates, it will reduce their temperature and relieve the danger.

It has been found by experiment that boiler flues that have been purposely allowed to become bare and overheated have been saved by suddenly directing a stream of cold feed water upon them, though such treatment may make them leak at the joints. The heat stored in such hot plates is insignificant as compared with the heat in the water and steam in the boiler.

Though violent explosions of new boilers in good condition have occurred from excessive pressure, they are very rare and can occur only as the result of criminal negligence; the boiler will usually give evidence of distress long before it is in danger of explosion. In case of over pressure, the same methods for checking the fire and reducing the steam pressure may be used, as are recommended for low water and overheating; if the pressure is so high that an explosion is to be expected, the chances of success are less and the risks are greater than with low water.

CHAPTER XI.

BOILER ACCESSORIES.

Safety Valves are intended to prevent the pressure of steam from rising to a dangerous point. In order to accomplish this, the effective opening of the valve should be sufficient to discharge all the steam that the boiler can make when urged to its full capacity. The effective opening is equal to the circumference of the valve seat multiplied by the lift of the valve, if the valve seat is flat; if the valve seat is conical, the lift should be measured at right angles to the seat. Then if l is the vertical lift and if α is the angle which the seat makes with the vertical, the effective lift is

$$l \sin \alpha.$$

The lift of a safety valve rarely exceeds $\frac{1}{10}$ of an inch. A two-inch pop safety valve made by the Crosby Gauge and Valve Co., was found by experiment to lift from 0.07 to 0.08 of an inch.[1] The valve had a conical seat with an angle of 45°. The actual flow was about 95 per cent of the calculated flow for this valve.

The amount of steam that a boiler can make may be estimated from the grate area, the rate of combustion, and the evaporation per pound of coal. The first item is fixed, and the other two, though somewhat indefinite, may be estimated from the type of boiler and the conditions under which it works.

For example, a factory boiler having a grate 5 feet by 6 feet may be assumed to burn 18 pounds of coal per square foot of grate surface per hour, and to evaporate 8 pounds of water per pound of coal. It will therefore generate

$$\frac{5 \times 6 \times 18 \times 8}{60 \times 60} = 1.2 \text{ pounds of steam per second.}$$

Napier's rule gives for the discharge of steam per second through an area of F square inches, and under the pressure of p pounds absolute,

$$G = F \frac{p}{70}.$$

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[1] Thesis by E. C. Burnham.

Here the weight of steam to be discharged per second is 1.2 pounds; consequently the area required, if the pressure is 100 pounds absolute, is

$$F = \frac{1.2 \times 70}{100} = 0.84 \text{ square inches.}$$

If the effective lift be assumed to be 0.075 of an inch, the circumference of the valve seat should be

$$0.84 \div 0.075 = 11.2 \text{ inches,}$$

and the diameter should be 3.5 inches.

A common rule requires that there shall be an area of $\frac{1}{3}$ of a square inch through the valve seat for each square foot of grate surface.

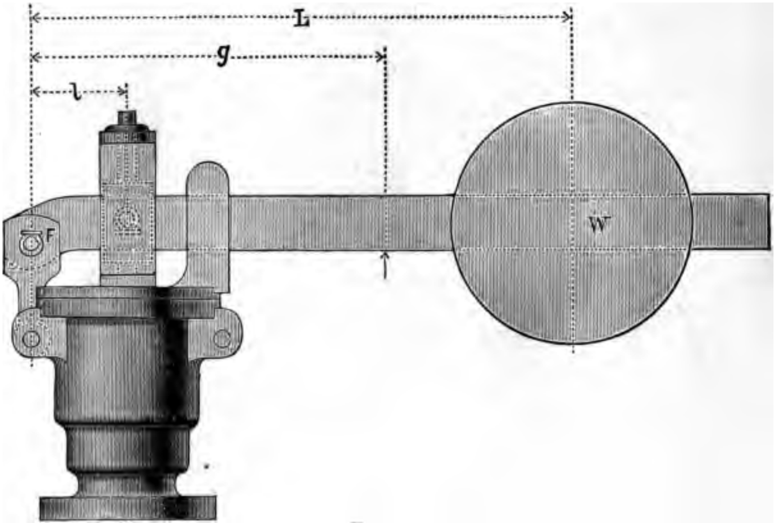

FIG. 66.

It so happens that this rule gives almost identically the same result as that just calculated for the above example; thus: —

$$\frac{5 \times 6}{3} = 10 \text{ square inches,}$$

$$\sqrt{\frac{4 \times 10}{\pi}} = 3.5 + \text{ inches diameter.}$$

Lever Safety Valve. — The external appearance of a well-made, *lever-loaded safety* valve is shown by Fig. 66. The bearings at the fulcrum and at the valve spindle are on knife edges to reduce friction: if pins are used instead, they must fit loosely to ensure that they will not stick even if rusted. The fulcrum is at the end of a small link to avoid cramping the valve spindle.

To calculate the weight to be applied to the lever, take moments about the fulcrum *F*. Thus, if the weights of the valve and of the lever are W_v and W_l, if the area of the valve is *A*, and the pressure of the steam above the atmosphere is *p*, then

$$pAl = WL + W_v l + W_l g,$$

from which *W* may be readily calculated.

Pop Safety Valve. — A defect of the common safety valve is that it does not close promptly when the steam pressure is reduced, and it is apt to leak after it has returned to its seat. A number of pop safety valves, so called from the suddenness with which they open and close, are in common use, in which prompt action is attained by causing the steam to increase the lifting force by reaction. The valve made by the Crosby Co. is shown by Fig. 67. *C* is the valve with a conical seat at *D*, and with a V-shaped lip shown at *F*. When the valve opens, the reaction of the steam against the V-shaped lip throws the valve wide open. The outer rim of the seat at *F* is perforated by numerous vertical holes not shown in the cut, by which the steam thrown down from the lip may escape, thus reducing the reaction; the ring *G* may be set to partially close these holes, and thus regulate the reaction and the rapidity of opening and closing.

For close regulation, it is desirable that the seat of the valve should be made narrow; in order that the valve and its seat shall endure the pressure thus concentrated on a narrow surface, they must be made of hard bronze or composition. Nickel alloys are sometimes used for this purpose.

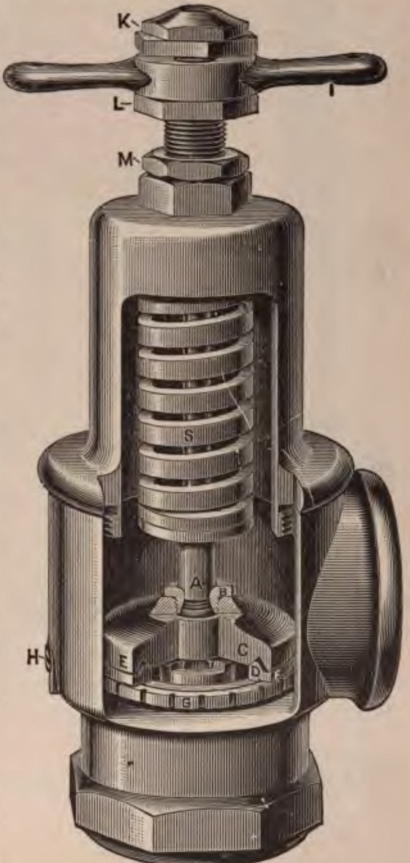

FIG. 67.

The spring is loaded with a helical spring, and the tension can be adjusted by the handle *I*, and set by the check nuts at *M*.

A safety valve should be so arranged that it can be opened by the engineer or fireman at will, and at least once a day it should be lifted from its seat momentarily, to be sure it is in working condition.

Safety valves are set to blow off at the desired pressure as compared with a steam gauge that is known to be correct.

To avoid the annoyance of blowing steam into the fire room, the discharge from the safety valve is often piped out of doors. There should be a drip to prevent the accumulation of water in such an escape pipe; in cold weather, if the fire is allowed to go out, the pipe may freeze up, should water be allowed to accumulate, and then the valve may fail to act.

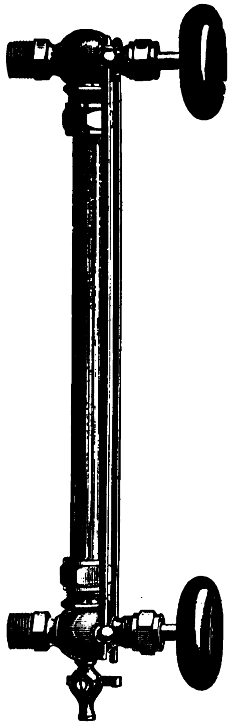

FIG. 68.

Water Glass. — The level of water in a steam boiler may be shown by a water glass like Fig. 68. Valves at the top and bottom allow the steam to be shut off if the glass breaks or needs cleaning, and a pet-cock at the bottom allows the water to be blown out of it. The brass fittings are, from preference, screwed directly into the boiler plates. When that cannot be done, the water glass may be put on a water column like that shown by Fig. 69, having a free communication with the steam and water space. The same cut shows three gauge cocks, one at the water level, one above, and one below. The steam discharged from the cock leading from the steam space is blue from superheating due to throttling; the mixture of water and steam from the cock leading from the water space has a distinctive white color. The level of the water may, therefore, be determined approximately by opening these gauge cocks.

The bursting of a gauge glass is both troublesome and dangerous. The device shown by Fig. 70 is designed to automatically shut off the steam in case of breakage. A small ball in the conical bore of the fitting rests against a wireguard, as shown, and does not interfere with the flow of water or steam under the usual working conditions. Should the glass break, the ball is thrown against the orifice by the rush of the water or steam, and is held there by the steam pressure. A pin on the end of the valve spindle throws the ball off when the valve is closed by hand.

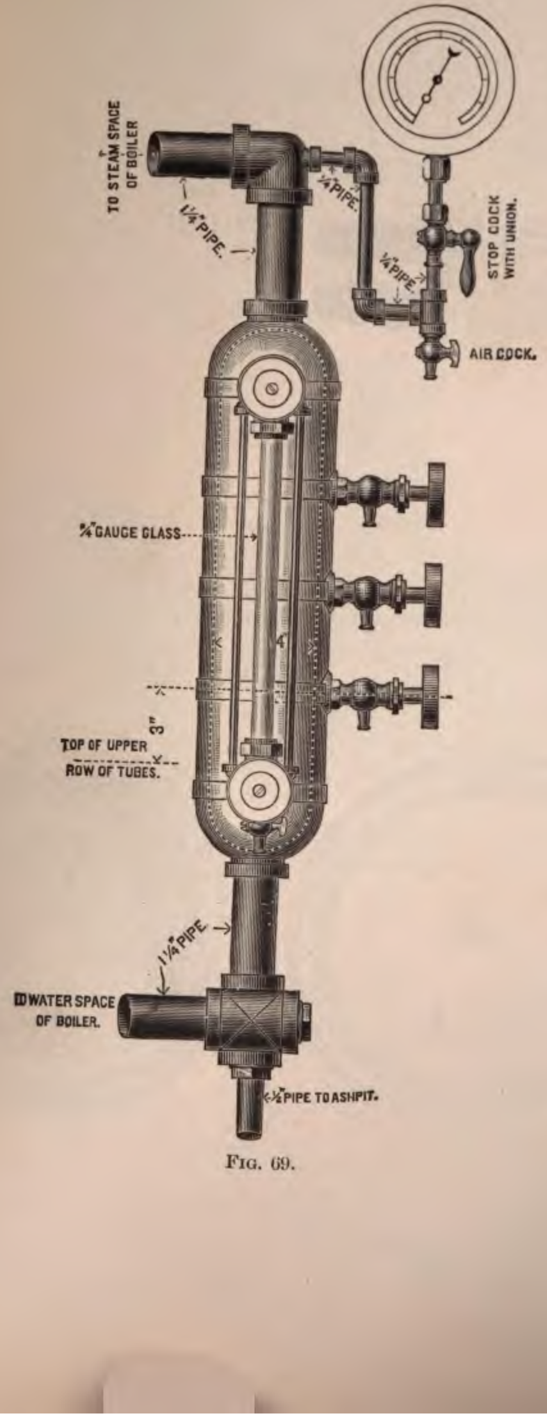
TO STEAM SPACE OF BOILER
1¼" PIPE.
¼" PIPE.
¼" PIPE.
STOP COCK WITH UNION.
AIR COCK.
¾" GAUGE GLASS
4"
3"
TOP OF UPPER ROW OF TUBES.
1¼" PIPE
TO WATER SPACE OF BOILER.
½" PIPE TO ASHPIT.

FIG. 69.

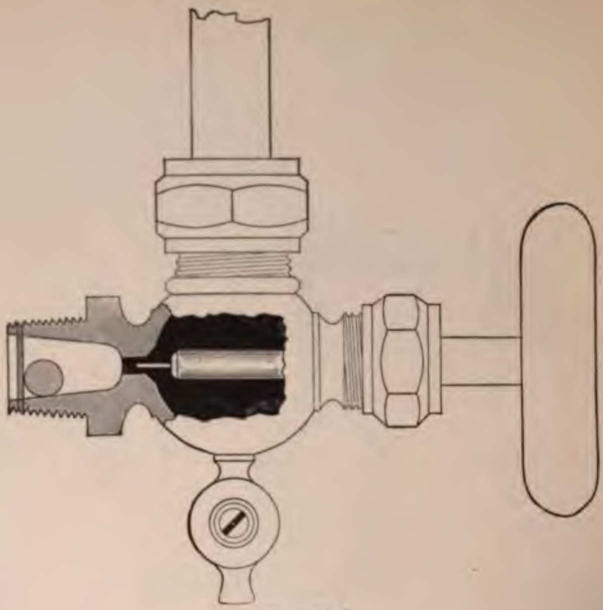

FIG. 70.

Steam Gauges. — The pressure of the steam is indicated by a spring gauge shown by Figs. 71 and 72. The essential part of the gauge is a flattened brass tube, bent to the arc of a circle. Pressure inside of the tube bulges out the flattened sides, and consequently the curvature of the tube decreases; the tendency is for the tube to become round and to straighten out. One end of the tube is fastened, and the other is connected by a multiplying device, to a needle that moves over a gradu-

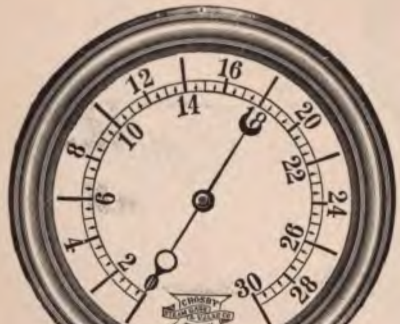

FIG. 71.

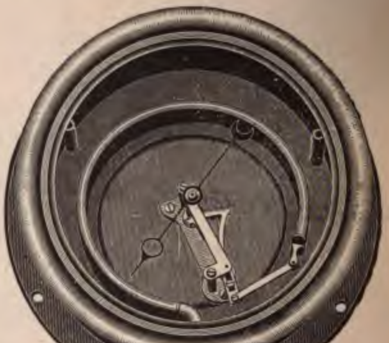

FIG. 72.

ated dial. The dial is graduated by trial to agree with the readings of a mercury column, or else with the readings of a standard gauge.

The long, flexible spring tube is liable to vibrate to an undue extent when the gauge is exposed to the jarring of a locomotive. To avoid this difficulty, two short stiffer tubes have their ends connected to a more effective multiplying device shown by Fig. 73. The greater number of joints in this device makes it less sensitive than the other form.

Since the spring tube changes its shape if the temperature changes, hot steam should not be allowed to enter it. An inverted syphon or U tube filled with water is, therefore, interposed between the gauge and the steam from the boiler.

Safety Plugs, such as shown by Fig. 74, are made of brass and provided with a core of fusible metal. If the plate into which they are screwed is in danger of overheating, the fusible metal will melt and run out, and steam and water will blow into the furnace. If the fire is not put out, it will at least be checked and the attention of the fireman will be attracted.

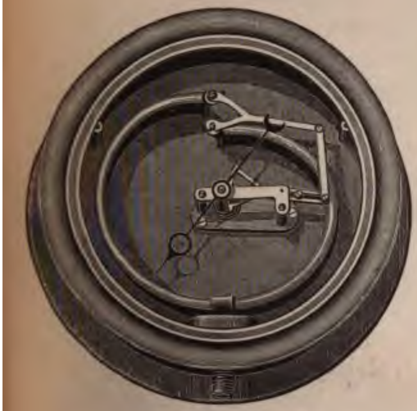

Fig. 73.

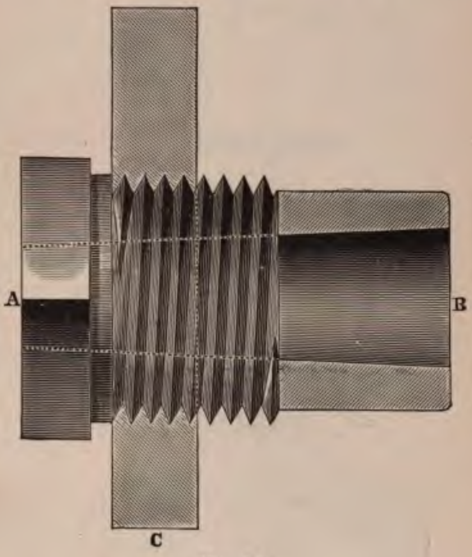

Fig. 74.

The melting point of fusible metals is not always certain, and the plugs not infrequently blow out when there is not apparent cause. On the other hand, they sometimes fail to act when the plate is overheated. If the plug is covered with incrustation, the fusible metal may run out without giving warning.

Low Water Alarm. — The device shown by Fig. 75 consists of two tubes in communication with the steam and water space of a boiler.

It is set at the water line, so that the upper tube is filled with steam, and has the corresponding temperature, while the lower tube is filled with water, which on account of the lack of circulation becomes cooled by radiation. Should the water line fall so that steam enters the lower tube, the expansion of that tube opens the valve of a steam whistle.

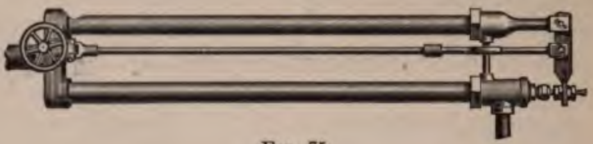
FIG. 75.

Reducing Valve. — When steam is desired at a less pressure than that of the boiler, it is passed through a reducing valve like that shown by Fig. 76. The valve *D* is held open by the spring at *MM*, until the steam pressure in the exit pipe *B*, pressing on the diaphragm *KK*, is able to overcome the spring and close the valve. The pressure at which this may occur is determined by the tension of the spring, which may be regulated by the screw under the head *N*. The hand wheel *R* allows the valve to be closed by hand. It is expected that the valve will be drawn down so as to admit just the proper amount of steam to the exit pipe *B*, to maintain the desired pressure in it. Valves for this purpose are liable to work intermittently, *i.e.* they close till the pressure falls below the proper point, then they open and raise the steam pressure above that point. The valve is a species of throttling valve, and therefore cannot be expected to remain tight. If the machinery supplied by the reducing valve is liable to be injured should the steam reach boiler pressure, there must be a stop valve beyond the reducing valve which must be closed when no steam is drawn, and must be used to regulate the supply of steam until the amount drawn exceeds the leakage of the reducing valve.

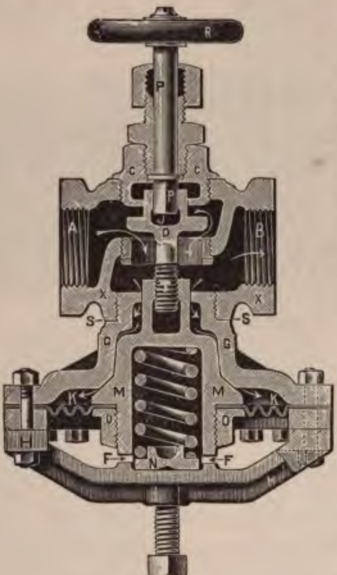

FIG. 76.

The Damper Regulator shown by Fig. 77 places the damper in the flue leading to the chimney under the control of the steam pressure, so that *if the pressure* of the steam falls, the damper is opened wider to quicken the fire. The pressure of the steam in the boiler is communicated through the pipe *a* to the lower surface of a diaphragm, and lifts the loaded lever *b*, which stands half way between the stops at the middle of its length, when the steam pressure is at the proper point. Should the steam pressure rise above the proper point, it raises the lever and opens a small piston valve at *c*, and water from a hydrant flows into *d* and presses on a piston which lifts the weights at *e* and so shuts the damper. The weighted head *e* of the piston is connected by a chain to the lever *d*, and closes the valve *c* as it rises, and so shuts off the water from the hydrant.

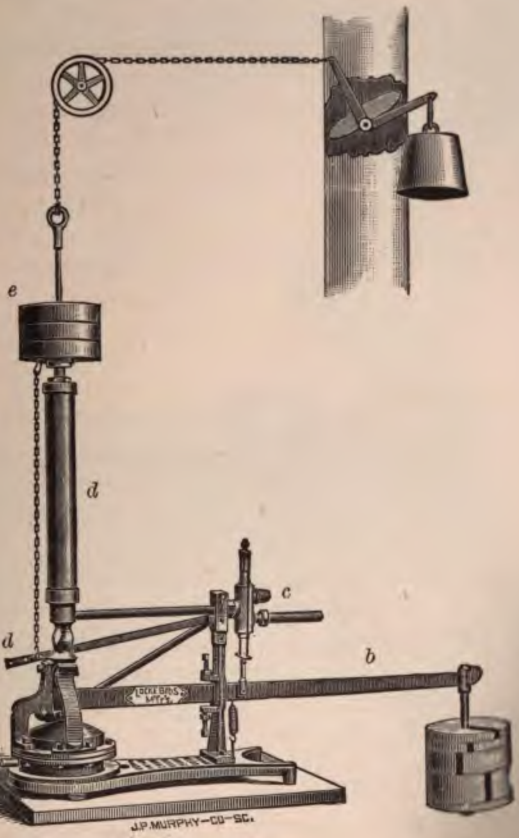

FIG. 77.

A regulator of the same form attached to a throttle valve acts as a reducing valve, and regulates the pressure below the valve with a variation of less than one pound. Fig. 78 shows the steam valve used when the Locke regulator acts as a reducing valve. The valve is a double valve which is nearly balanced, but with a slight tendency to rise under steam pressure, as the lower valve is the larger. The cylindrical part of the valve is cut into V notches, so that the supply of steam is regulated to a nicety when the valve is partially open. The cylindrical portion of the valve protects the valve seat and the valve face so that the valve may remain tight when closed.

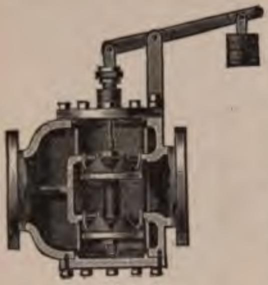
FIG. 78.

Steam Traps are intended to drain water from steam pipes without allowing the steam to escape. Fig. 79 represents a trap with a float which rises as the water accumulates and opens the out-flow valve at the left, so that the water may flow out under the pressure of the steam; but the float falls as the trap is emptied, and closes the out-flow valve before it is exposed to the steam, which, therefore, cannot escape.

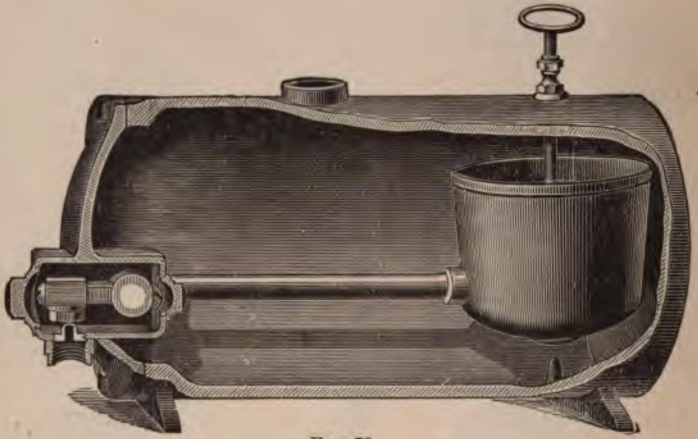
FIG. 79.

Expansion traps are convenient to free pipes from water that may accumulate while the pipe is cold. The essential part is a long rod that contracts when cold, opens a valve, and allows the cold water to flow out; but when the pipe is again filled with hot steam, this follows the cold water through the trap, expands the rod, and closes the valve again.

Globe Valves are named from the globular form of their cases. The case is separated into two parts by a diaphragm with a passage through its horizontal part, as shown in Fig. 80. The fluid enters at the right, passes under the valve, and out at the left. The valve is shut by screwing down the handle on the valve spindle. A stuffing box around the valve spindle prevents leakage of fluid. In this valve the seat is *rounded*, and the valve face is a ring of a peculiar composition, let into the valve at *R*. When the valve is shut, this composition is squeezed down onto the seat and makes a tight joint.

If the fluid enters the valve as shown, the valve spindle may readily be packed to prevent leakage, while the valve is closed. If the fluid entered the valve at the other end, it would be necessary to shut off the fluid from the entire pipe in order to pack the valve.

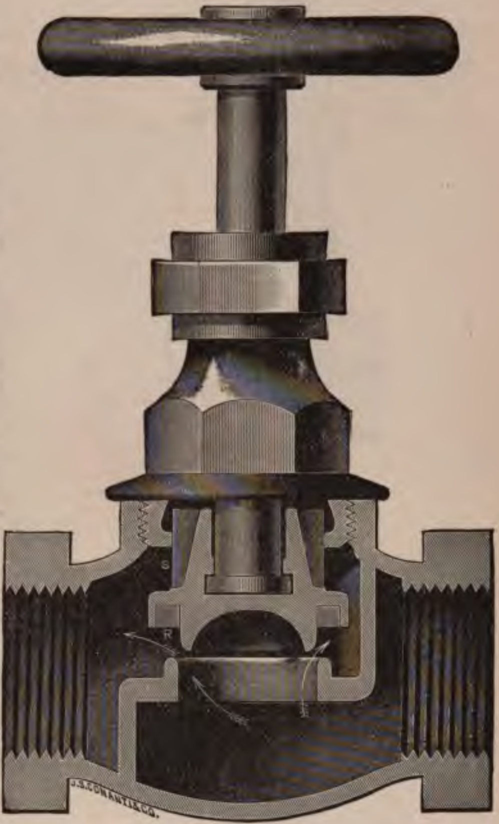

FIG. 80.

Gate Valves give a clear, direct passage through the valve when open. Figs. 81 and 82 show a gate valve which is opened by a lever; the lever, however, is not shown in the cuts.

Check Valves allow the passage of the fluid in one direction only, as shown by Fig. 83. A check valve is always placed between the boiler and a feed pump, so that the water from the boiler cannot flow out through the pump when the pump is at rest, even though the pump valves may leak. A stop valve should be placed between the boiler and the check valve in order that the latter may be examined when the boiler is working.

Feed-Water Heaters. — The feed water supplied to a boiler may be heated up to the temperature of the exhaust steam by passing it through a feed-water heater. Feed-water heaters are sometimes made open, *i.e.* the steam from the engine mingles with and heats the feed water. Such a heater arranged for throwing down carbonate of lime from water is shown by Fig. 25, page 39. Such heaters have the disadvantage that the oil from the engine is carried into the boiler.

A closed feed-water heater resembles a surface condenser, and as the steam and water do not mingle, there is no danger of carrying oil from the engine into the boiler. The Wainwright heater, shown

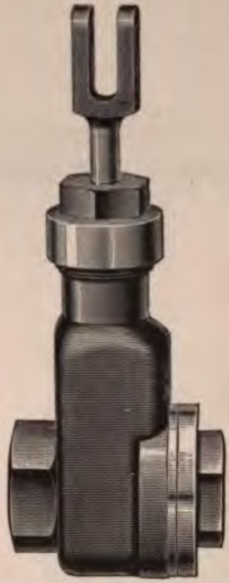

FIG. 81.

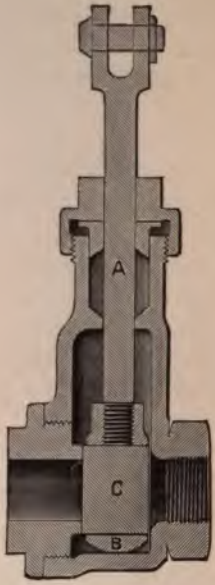

FIG. 82.

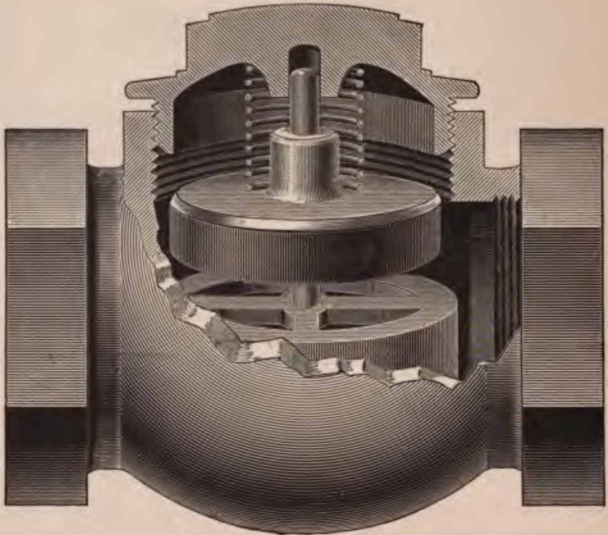

FIG. 83.

by Fig. 84, has the heating surface of corrugated copper or brass tubes, of their peculiar make, to allow for expansion. The steam from the engine passes around the tubes and the feed water passes through the tubes.

The Berryman feed-water heater, shown by Fig. 85, is arranged to have the exhaust steam pass through a series of inverted U tubes, around which the feed water circulates.

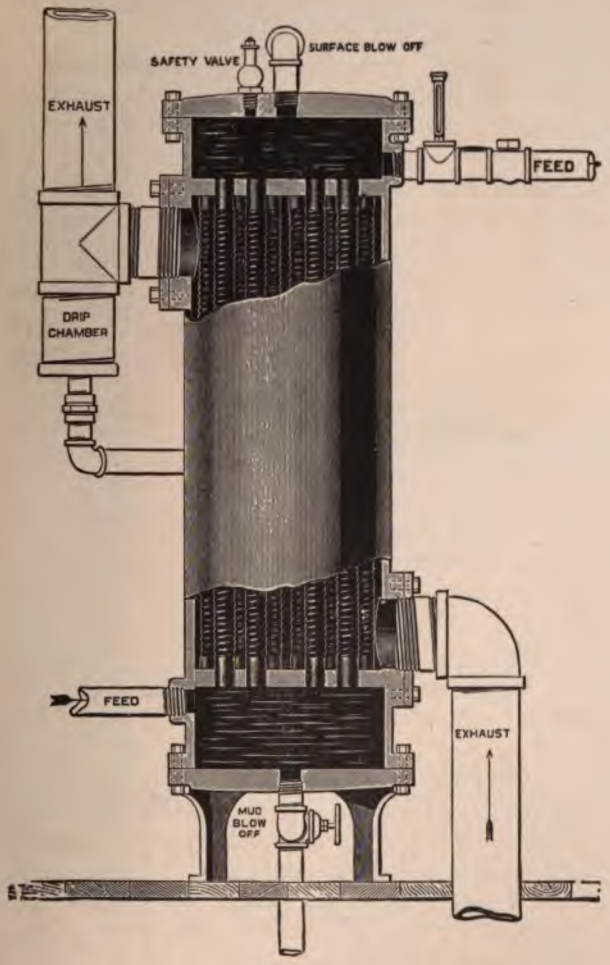

Fig. 84.

Live steam feed-water heaters take steam from the boiler to raise the temperature of the feed water up to, or nearly to, the temperature in the boiler. The principal advantage appears to be that unequal contraction, due to the introduction of cold feed water, is avoided. It is claimed that for some forms of boilers a better circulation is obtained by aid of such a heater.

Feed Pipes. — The temperature of the feed water is usually much below the temperature in the boiler. It thus becomes essential to so locate the inlet, and to so distribute the water, that undue local contractions may not occur; this is of special importance when the supply is intermittent. The feed pipe for the cylindrical tubular boiler, shown by Plate I., enters the shell near the water line, through a branched and perforated pipe.

Fig. 86 shows a device for mingling the feed water with the steam in the steam space of a boiler, so that it may be heated before it can come in contact with the tubes or other parts of the boiler.

The feed pipe is sometimes carried through the steam space as shown by Fig. 87, in which the feed-water pipe passes twice through the length of the boiler before it enters the mud drum, where the suspended impurities and the lime salts thrown down by heat are expected to be deposited. There are evident objections to placing the feed pipe in an inaccessible place, which become enhanced if the pipe is liable to be choked with deposits. In this example the pipe was in one of four two-flue boilers shown by Fig. 88, which was joined to a mud drum at the back end and to a steam drum at the front end; and difficulty arose from unequal expansion and corrosion in the mud drum near the end of the feed pipe.

Feed Pumps. — Boilers are commonly fed by small direct-acting steam pumps placed in the boiler room. The steam consumption per horsepower per hour of such pumps is very large, and yet the total steam used is insignificant. They are cheap and effective, and easily regulated.

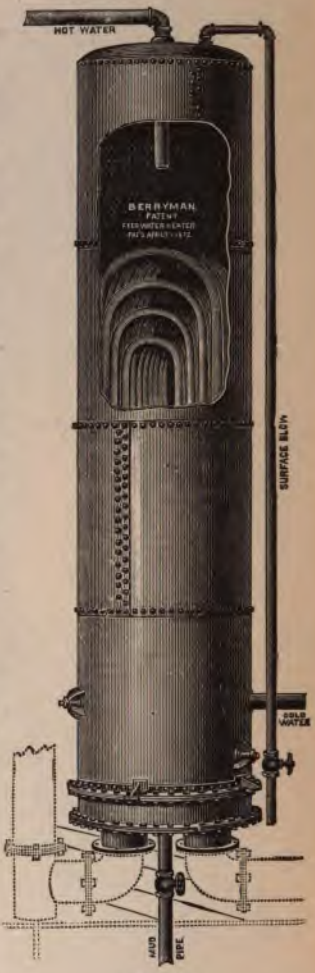

FIG. 85.

Power pumps driven from a large engine are more economical, *provided their* speed can be regulated; they not unfrequently are arranged to pump a larger quantity than required for feeding the boiler, the excess being allowed to flow back to the suction side of the pump through a relief valve.

When one pump supplies several boilers, a series of difficulties are liable to arise. First, if the boilers are fed singly in rotation, the large intermittent supply of feed water is likely to give rise to local contraction, and the water level in the boiler fluctuates; there is liability that the water level will fall too low, endangering the heating surface, or there may be excessive priming when the water level is high. It appears advisable that the feed should be delivered to all the boilers simultaneously, the supply to each boiler being regulated by the stop valve; each branch pipe to a particular boiler should be provided with its own check valve, and the water level and rate of feeding of each boiler must be carefully watched by the fireman, or, better, by a water tender, if there are many boilers.

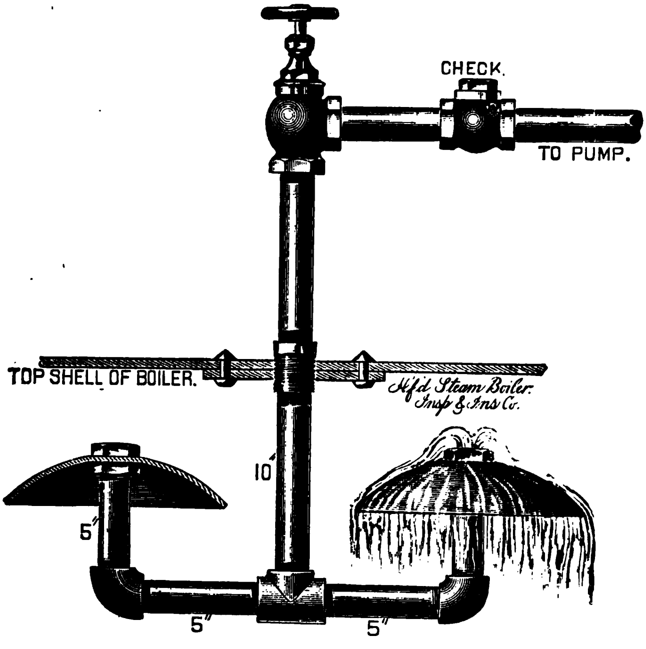

FIG. 86.

An injector is conveniently used for feeding a boiler if the feed water is not too hot; it has the incidental advantage that it heats the water as it feeds it into the boiler. An injector should be connected up with unions, so that it may readily be taken down for inspection.

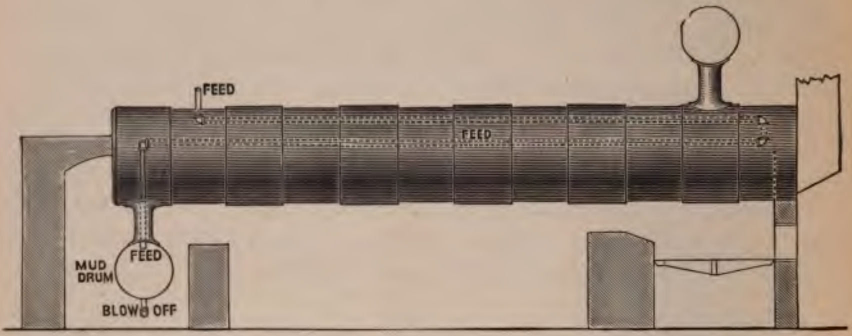

FIG. 87.

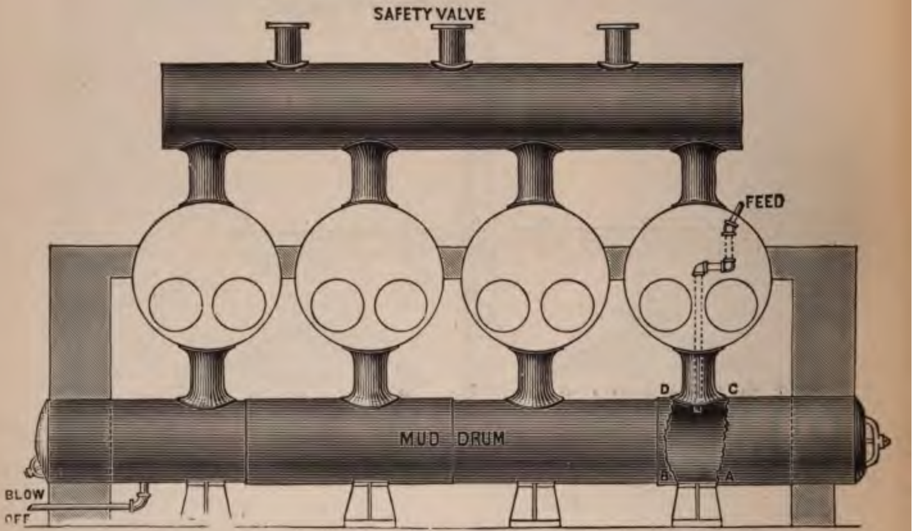

FIG. 88.

At sea an injector is commonly used when the boilers are fed from the sea or from a supply tank.

Blow-off Pipe. — The blow-off pipe draws from the lowest part of *the* boiler, or from some place where sediment may be expected to collect. It usually has a cock and not a valve, since leakage through a cock is likely to be noticed at the ends of the cone through which the passage is carried. The pipe should be carried beyond the cock, so that the attendant is not liable to be splashed with hot water, but the pipe should end in the boiler room or where discharge through the pipe on account of a leaky cock or valve may be sure to attract attention. Each individual boiler should have its own blow-off pipe. Figure 89 shows a blow-off pipe with a removable covering to facilitate the discovery of leaks.

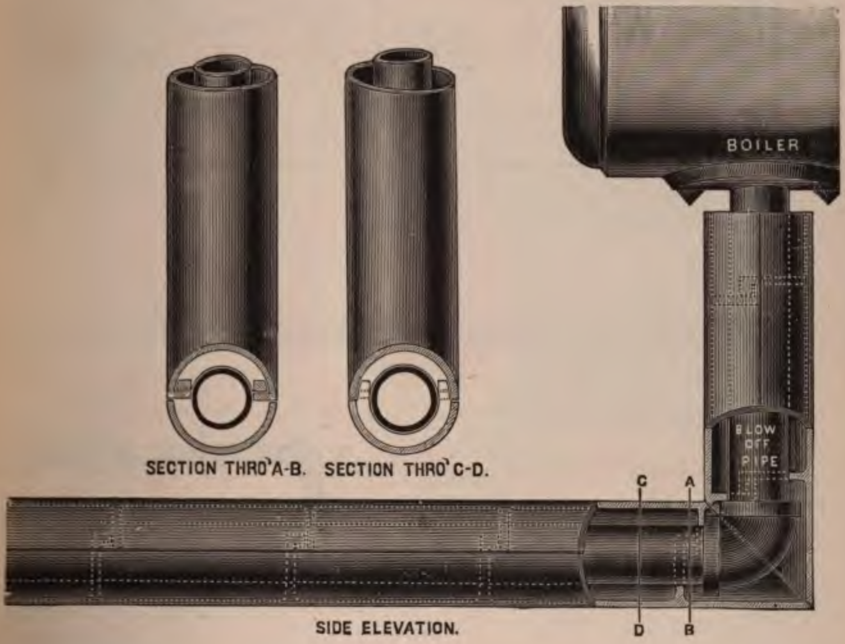

Fig. 89.

Piping to carry steam from a boiler to an engine, for heating buildings and for other purposes, is too important to be considered as a near accessory to the boiler. A few remarks may not be out of place, however.

The expansion of the pipe due to changes of temperature should be provided for, or else cracks in the pipe or fittings, or leakage at the joints may be expected. An example of a failure to attend to this matter is shown by Fig. 90, and Fig. 91 shows how the difficulty may be avoided by giving the pipe a chance to spring. *AE* is the steam nozzle on the boiler, and *D* is the steam main. There is a stop valve at *C* which makes it easy to shut down the boiler when desired.

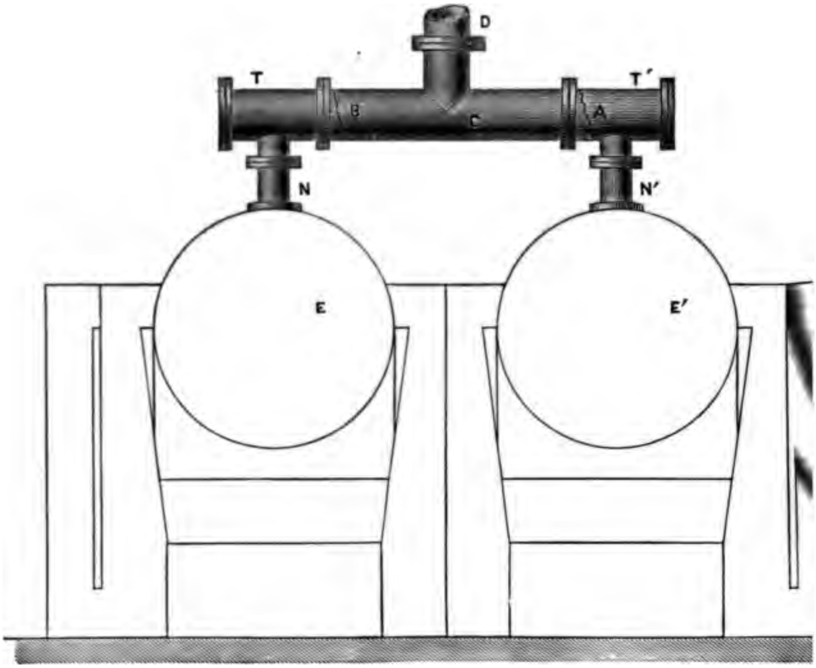

FIG. 90.

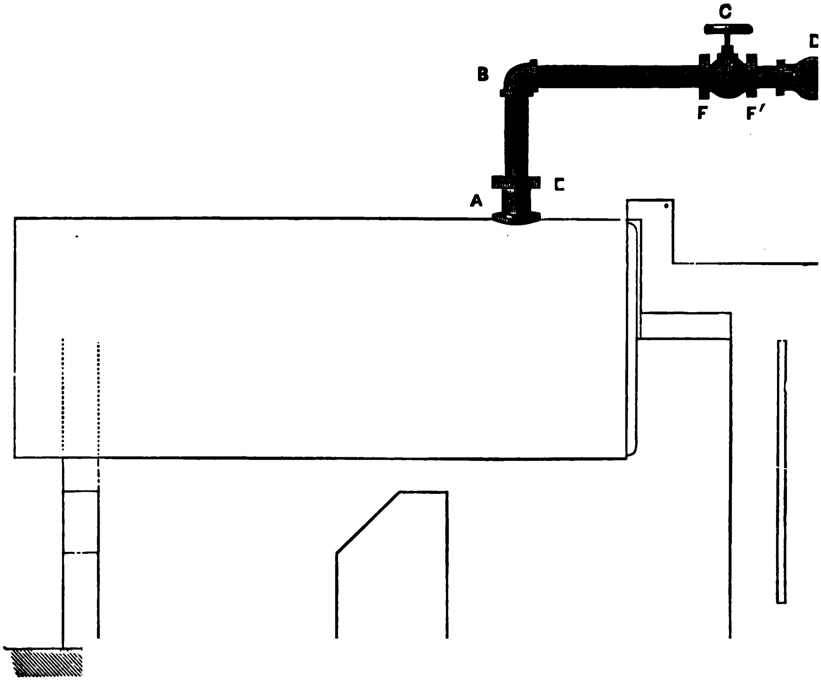

FIG. 91.

If pipes are liable to fill up with water when not in use, they ought to be provided with drips through which the water may be blown out; in some cases an expansion trap may be used to advantage.

Horizontal pipes are frequently arranged so that water may collect in them, due to a sag in the pipe or to the fact that they do not properly drain through a side branch, as shown by Fig. 92. Though the water may lie quiet in such a pocket while the draught of steam is steady, a sudden increase in the velocity of the steam, or a rapid opening of the valve supplying steam to the pipe, will sweep the water up and carry it along with the steam. The danger from the inrush of water to an engine is readily seen, but it is not so well known that the water thus violently thrown against elbows and other fittings gives rise to leaks, if it does not burst the fittings. It is to be remembered that steam offers little or no resistance to the movement of water in a pipe, as it is readily condensed either from a slight increase of pressure or by mingling with colder water. Again, water at the temperature corresponding with the pressure easily separates, forming bubbles of steam, which as easily collapse, and the shock of impact of the water gives rise to pressures that search out all weak places in the pipe, even at some distance. Figure 93 is a method suggested by the Hartford Boiler Insurance Company to ensure the drainage of a steam main by a branch pipe. They also suggest that

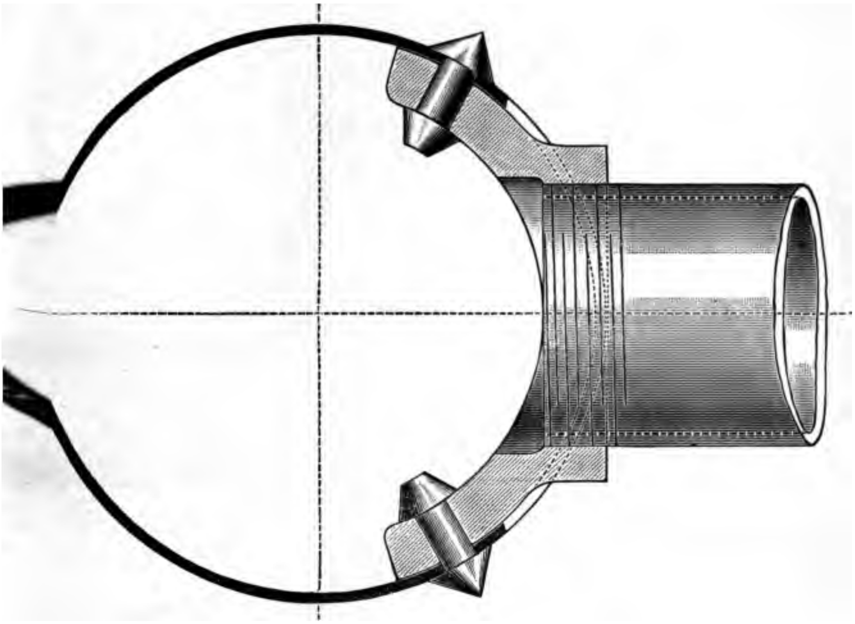

FIG. 92.

fittings of cast iron be similarly made. Of course the same object may be accomplished by drawing steam from the bottom of the main pipe.

Pipe Hangers. — Large pipes are commonly hung from overhead beams, as shown by Fig. 94; the dotted lines show how the pipe may be distorted if proper allowance for expansion is not made.

A device for a pipe hanger, with a roller to provide for a large amount of expansion, is shown by Fig. 95.

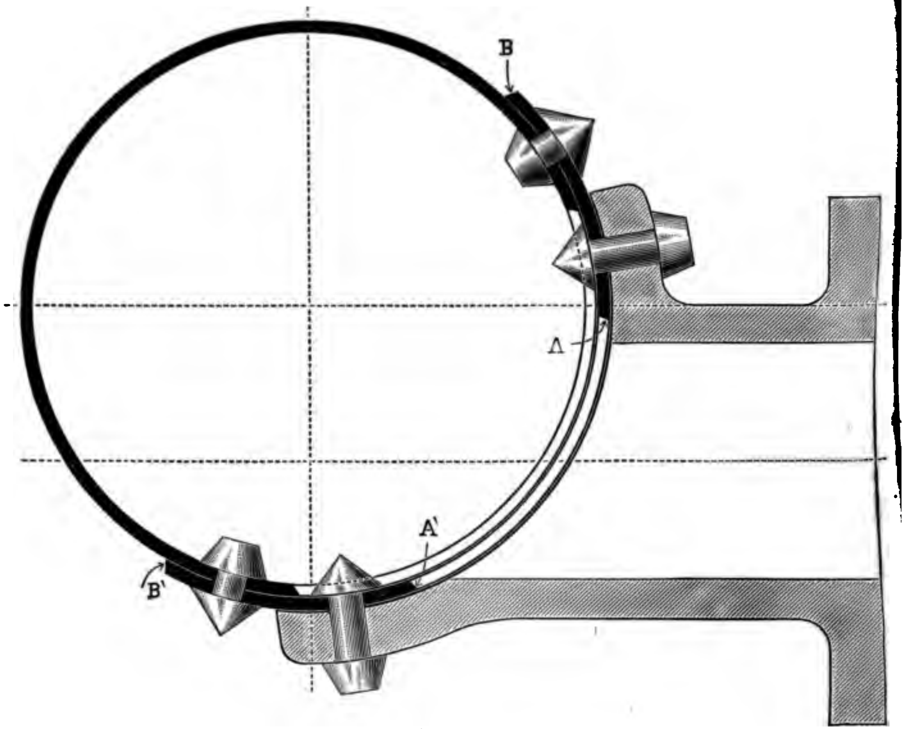

FIG. 93.

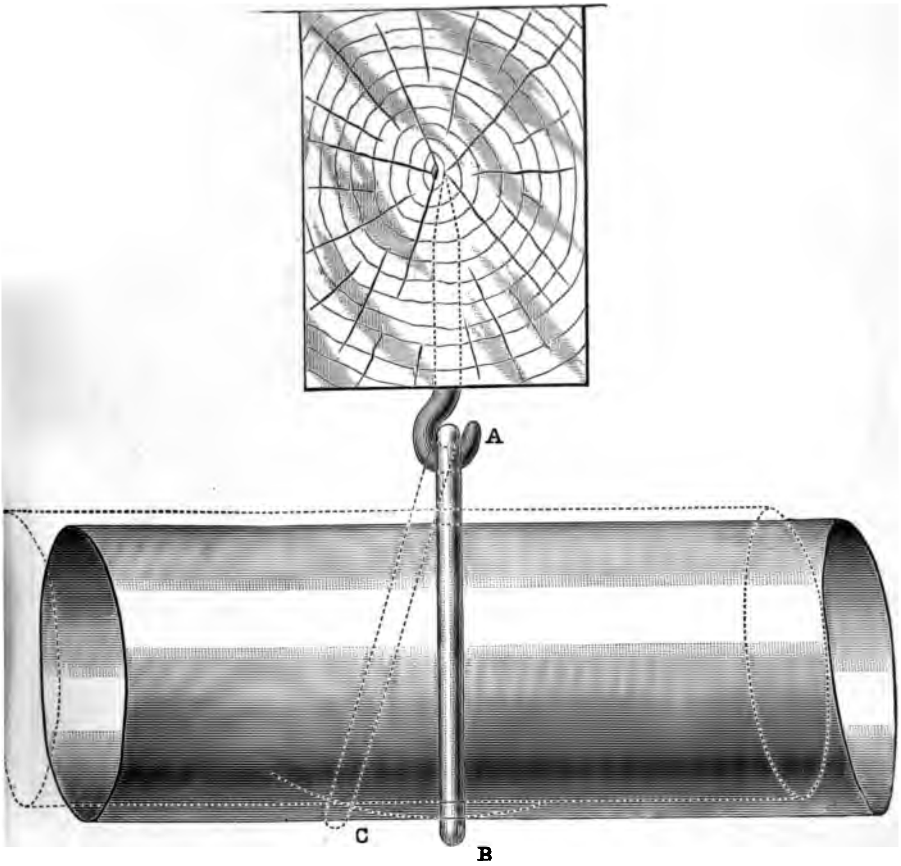

FIG. 94.

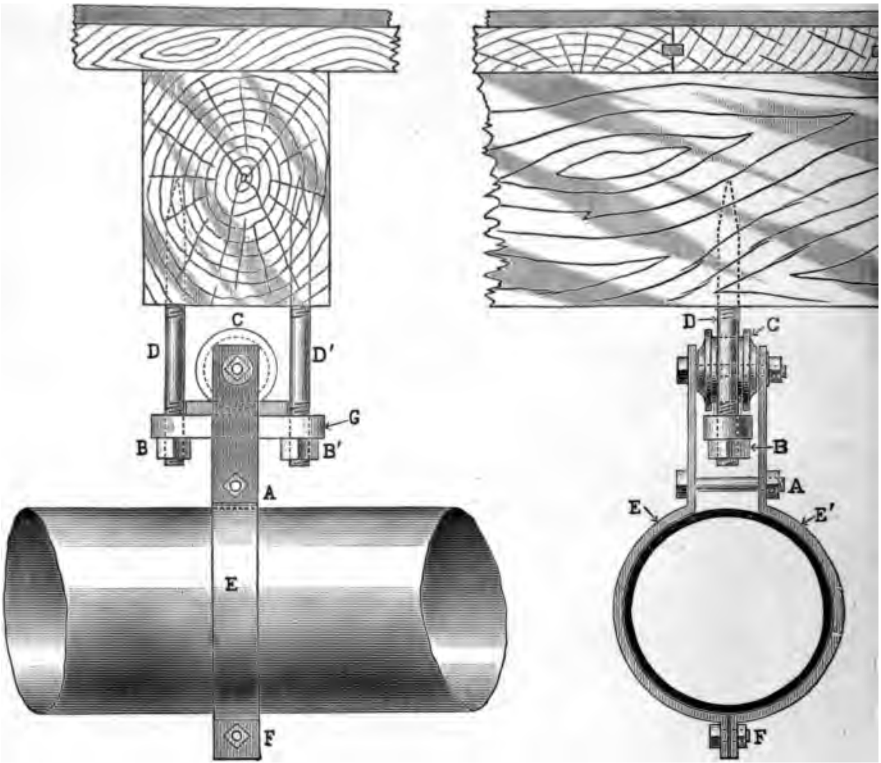

FIG. 95.

CHAPTER XII.

SHOP PRACTICE.

MODERN boiler shops have the tools so arranged that the material starting from the bench of the layer out travels forward, from machine to machine, until it reaches the riveter. It is advisable to have a separate swing crane over each machine; the cranes overlapping so that one crane takes the work from the next, and so on. The layer out should have his bench near the place where new plates are unloaded into the shop from teams. Near the riveter there should be a vacant space to allow of bolting the plates together preparatory to riveting, and beyond the riveter there should be considerable space in which to locate boilers while the tubes, stay rods, etc., are being put in.

A few machine tools are needed to do the necessary machine work on cast-iron fittings such as steam nozzles, manhole rings, covers, etc.

For a shop employing from 40 to 60 hands, two planers, three lathes, and three upright drills would be sufficient. Flanging and blacksmith work are generally done in a separate room. A pattern shop and a storeroom for patterns of grates, fronts, fittings, etc., is generally located in the second story.

Flanging. — Boiler shops which build regular sizes of horizontal boilers generally buy their boiler heads all flanged, but shops which do general work often have to make irregular sizes and shapes, and in such cases do their own flanging. The heads which are sold ready made are pressed into a former by means of a die, worked by a hydraulic press. As the whole head is heated, there are no strains set up by this method of flanging. When flanging is done in the shop, it is usually done by bending small portions of the circumference at a time over an anvil of cast iron.

This anvil or former, as it is called, is simply a block of cast iron about six inches deep, flat on top, with one side rounded at about the same radius as the boiler head. The corners are rounded so as not to cut the plate. The former is placed near a low forge used for this work.

In flanging, the plate is first marked at short distances on the inner circle of the bend with a prick punch. A portion of the plate is then heated to a good heat, removed from the fire, the coke being brushed off with a broom, and taken to the former. After adjusting so that the depth of flange required overhangs the right distance from the edge of the former, the heated portion of the plate is beaten down against the side of the former by wooden mauls and then smoothed with a flatter and sledge. The plate is then heated in a new place and another portion bent. To straighten the head and also to remove the strains set up by this way of flanging, it should be heated to a dull red and allowed to cool gradually.

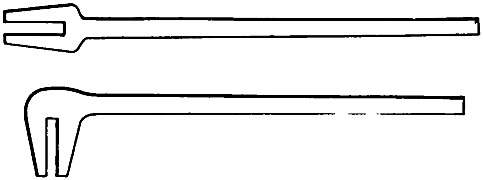

Fig. 96.—Lifting Dogs.

For convenience in handling plates these tools are used, Figs. 96, 97.

After flanging, holes for rivets, stay rivets, and tubes are marked and the rivet holes punched. To punch on the flange, a horizontal punch similar to Fig. 99 is used.

The punch shown in Fig. 99 will punch a 2″ hole in a 1″ plate. The depth of throat is 60 ″, and the distance from centre of punch to edge of frame is $2\frac{1}{8}''$. It is started and stopped by a treadle, and the plunger, which is of wrought iron, is operated by a cam and lever with a positive return motion. An automatic stop is provided, by which the return movement of the plunger may be arrested at any point of its stroke. The stripper is adjustable to any thickness of plate. The punches are held in position by a threaded cap of wrought iron, which also affords an adjustment for wear of the punches, as shown by Fig. 98.

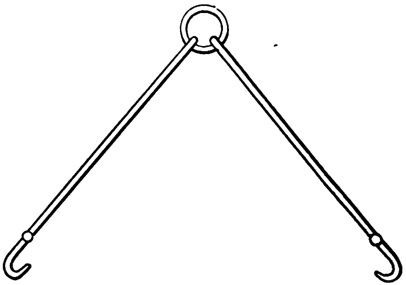

Fig. 97.—Crane Lifts.

Holes for the tubes are drilled under an upright drill made especially for this use. A drill shaped like Fig. 100 is used, the centre running in small holes previously made, which serve to guide the drill.

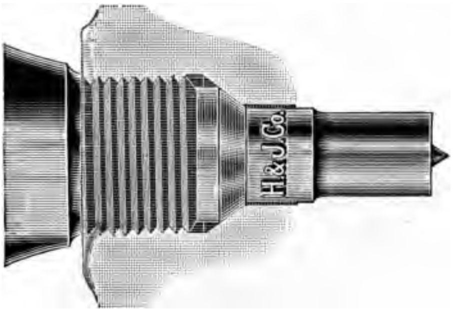

Fig. 98.

Tube holes are often punched out by what is called a spiral punch, Fig. 101. The acting face of the punch is really a helical surface which acts like the knife of a shears. The holes are afterward reamed out.

Lastly, the head is rotated by the platen of a boring mill, and the edge cut and bevelled for caulking. Plate VI. shows an 84″ boring and turning machine such as is *used for this purpose.*

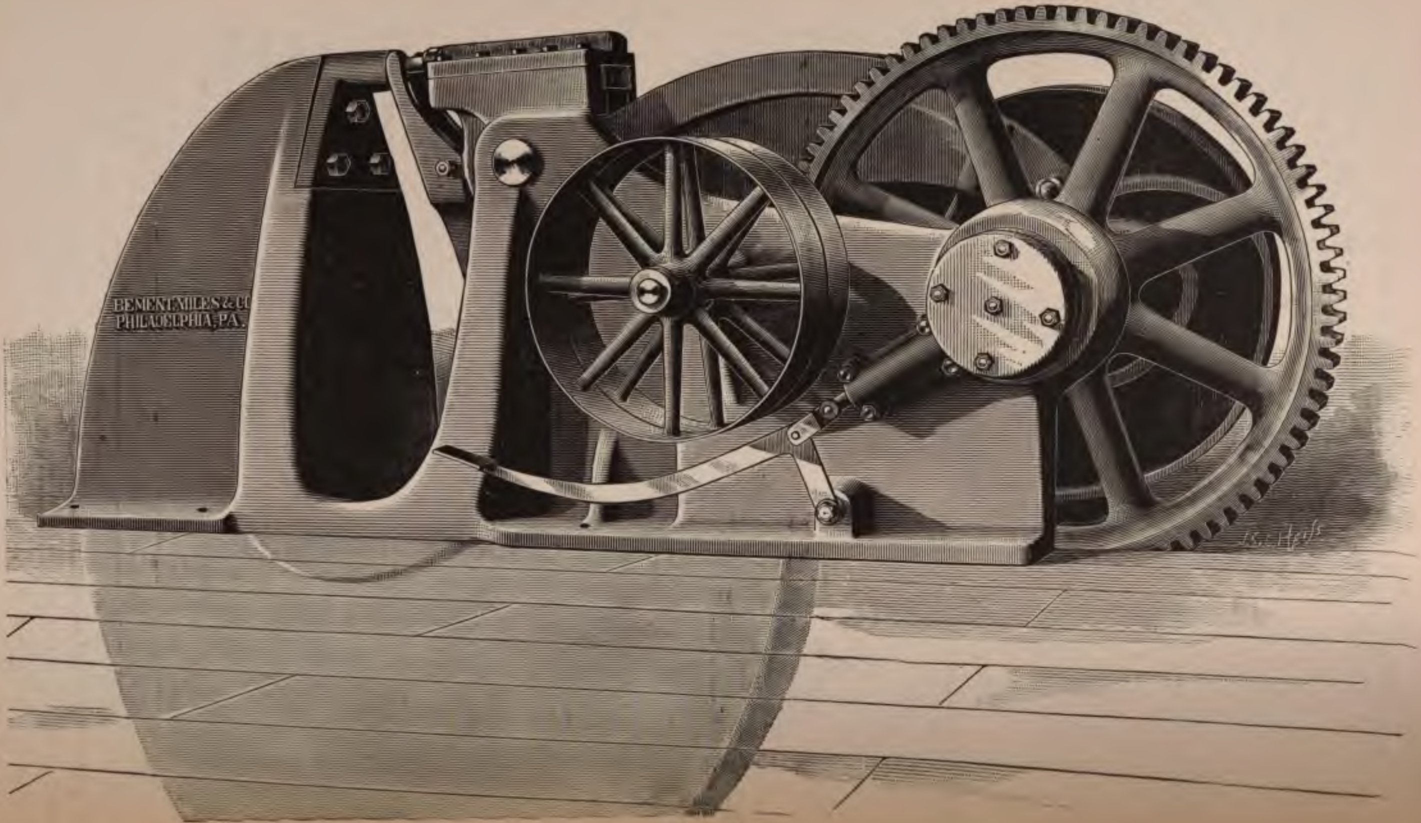

FIG. 99.

The first, and one of the most important steps in the work on the shell, is the marking out of the sheets. Generally one man in each shop does all the laying out. After squaring the sheet, he marks off the length and locates the rivet holes by means of gauges. These gauges have to be made by trial, a suitable allowance being made in them on account of the thickness of the plate for the change in length due to rolling. There is a gauge for each course, or a set of gauges for each size boiler, and also sets for the same size, but with different thickness of shell. The plates are marked either with a piece of soapstone or with a slate pencil. Rivet holes are prick punched at the centre.

The plate is now taken up from the bench by a crane, and conveyed to the shears, where any superfluous stock is removed.

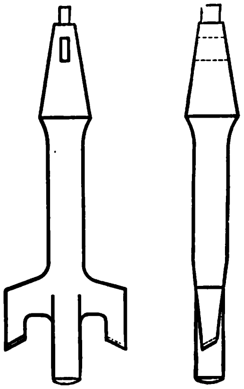

FIG. 100.

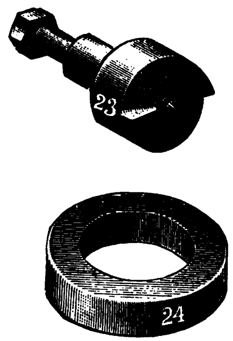

FIG. 101.

Figure 102 shows a shearing machine with crane attached. The lower knife of the shear is fixed. The upper knife is forced down by an eccentric inside of the head 3.

The eccentric shaft is coupled to the gear by means of a clutch thrown into action by the treadle. The weight of the sliding head is counterbalanced by a weight and lever at the top. The throat is 48″ deep; or, in other words, a cut can be made 48″ from the edge of the plate.

When shearing, bar-iron bolts are put through lugs on the side of the casting, so as to avoid straining the frame at the back of the throat. In both punches and shears, heavy fly wheels or balance wheels have to be depended upon to do a large part of the work. The energy stored in them during the up stroke is given out on the down or working stroke.

A heavy shearing machine driven by a separate engine is shown by *Plate VII.* It has a 36″ throat and can shear up to 1½″ plate.

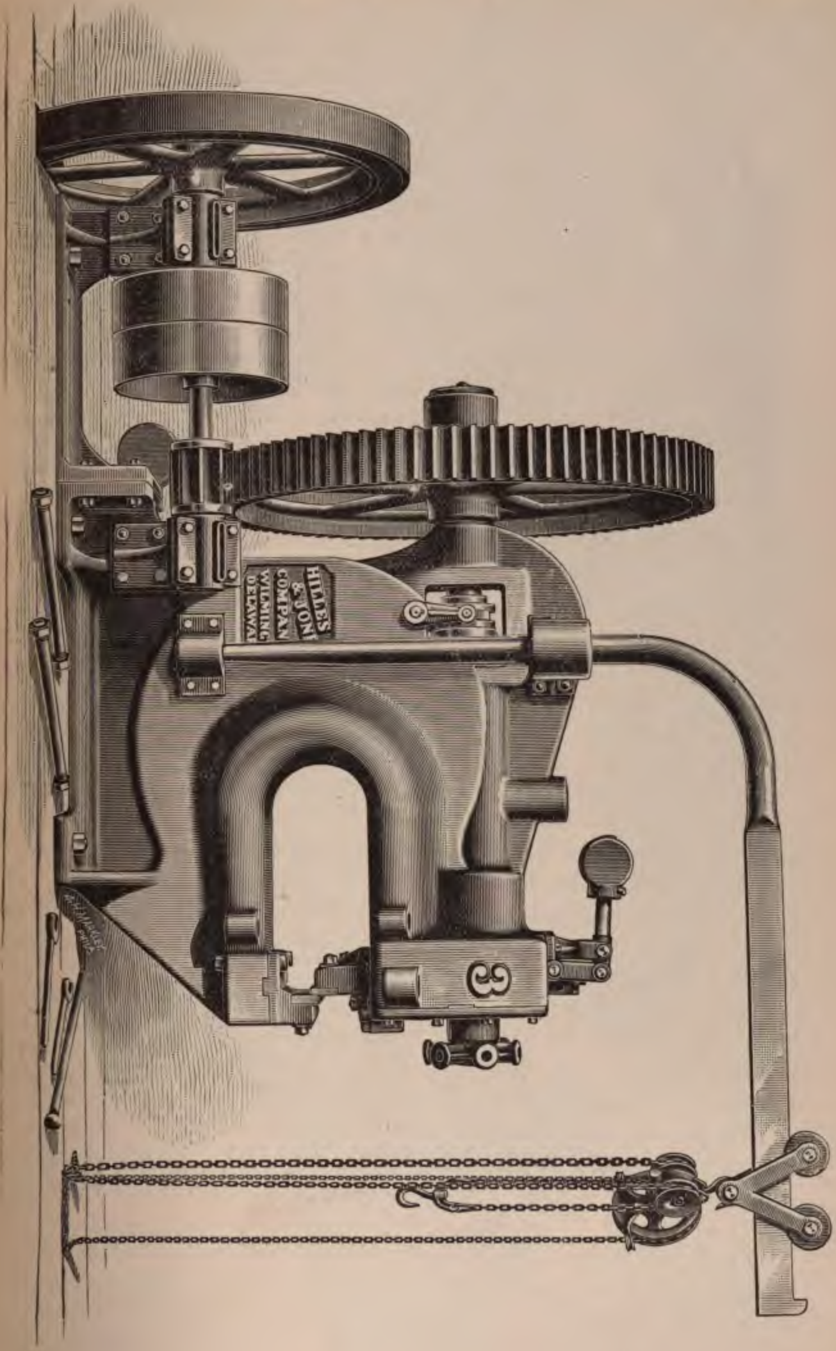

Fig. 102.

After leaving the shears the plate is moved to the punch, Plate VIII., where the rivet holes are punched, and also any large holes such as the openings for manhole rings or steam nozzles.

These holes are cut out by punching a series of small holes along the edge of the larger holes. To aid the workman in finding the punch marks at the rivet holes, there is a feeler or tit on the bottom side of the punch which fits into the prick-punch marks when the plate is in the right place (Fig. 103).

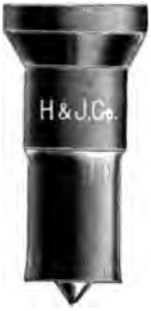

FIG. 103.

The plate is next taken to the plate planer (Figs. 104, 105, 106), and planed to a slight angle so as to give a good caulking edge. The plate is fastened in the planer by screws running down through the frame, or by jacks.

The tool is made double so as to cut on both strokes, and is moved forward and back by means of a screw driven alternately by an open and a crossed belt. The tool is fed in by means of a screw and wheel on the tool carriage. The plate is now heated and scarfed, or drawn down like a wedge, so that a tight joint can be made in the roundabout seam where the three thicknesses come together.

It is next rolled to the required shape or bent into a circle by being drawn back and forth between three rolls, the top one of which is forced downward by screws, as the radius of the bend becomes smaller. The plate is bent cold, and is of course passed lengthwise through the rolls.

The rolls, which are driven through gears by open and crossed belts, are reversed before the plate runs out. As the bottom or fixed rolls are some little distance apart, there is a short distance at each end of the plate which is not bent by the rolls. This is generally bent by sledges, but in some rolls there is a separate attachment on the end of the roll housing, by means of which the ends of the plate are bent before passing through the long rolls.

This attachment consists of two short rolls about 15″ long, one being concave and the other convex. The ends of the plate are fed through sideways, or at right angles to the direction in which it is afterward passed through the long rolls.

The rolls have to be as long as the greatest width of plate used. Oftentimes the top roll is counterbalanced so that it may be easily raised out of the way to enable a plate to be inserted or to be removed after bending.

Figure 107 shows a common form of horizontal roll. The two lower rolls are driven in either direction by heavy pinions keyed to the rolls outside the housing. The ends of the top roll can be drawn down either together or separately. The upper roll is extended, and by means of a screw and lever the left-hand end of the roll can be raised, the roll

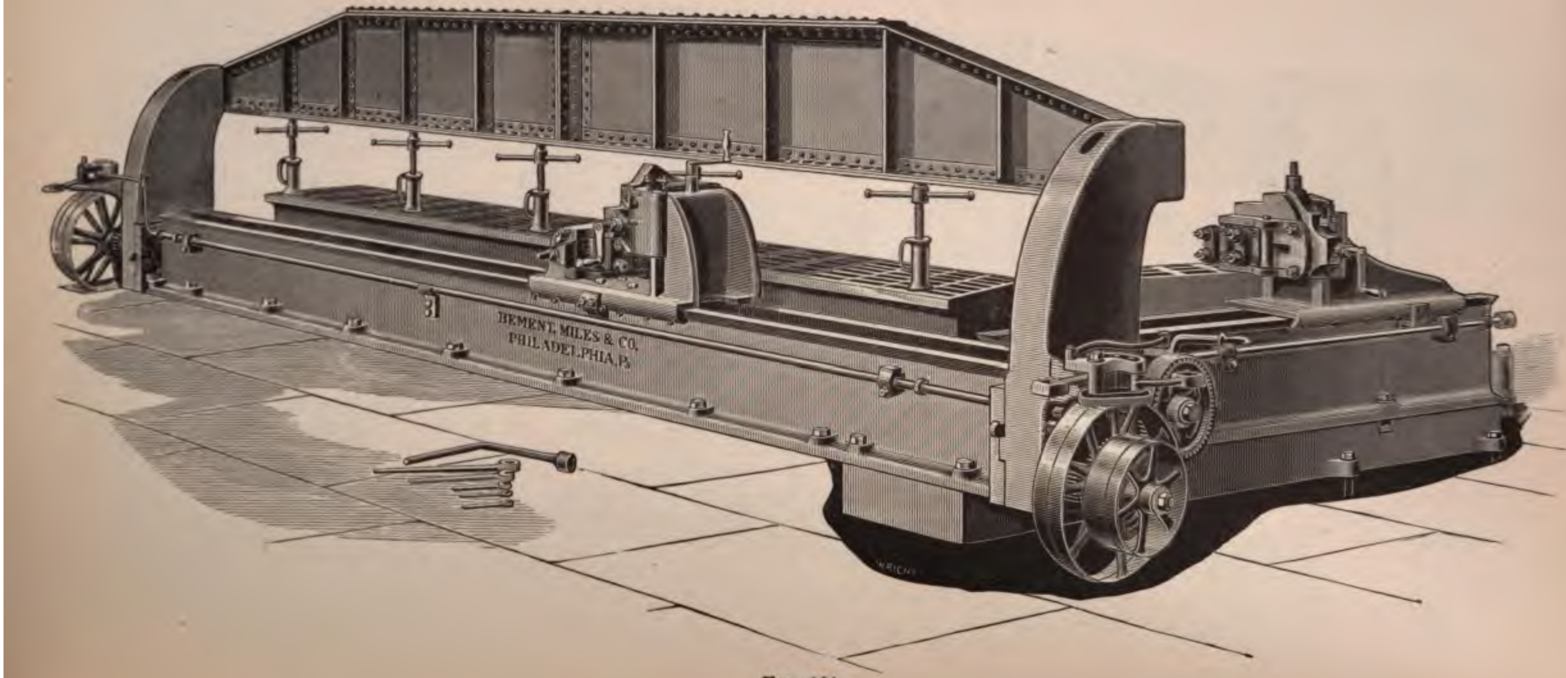
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Fig. 104.

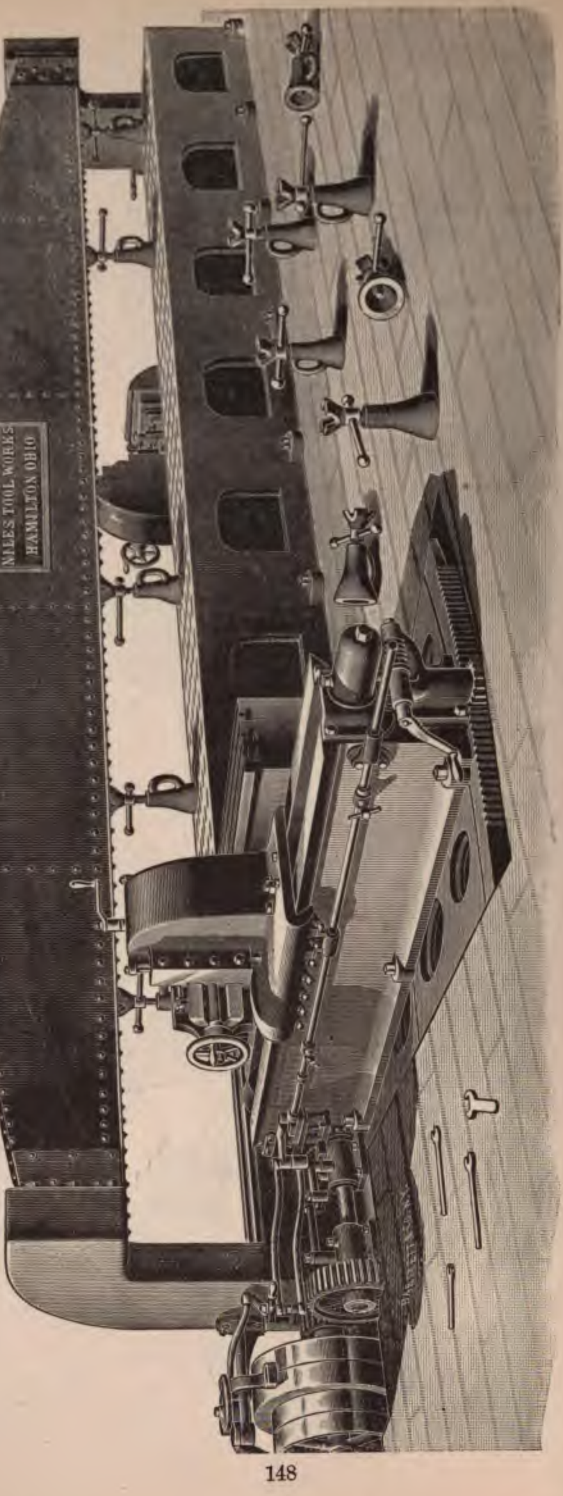

FIG. 105.

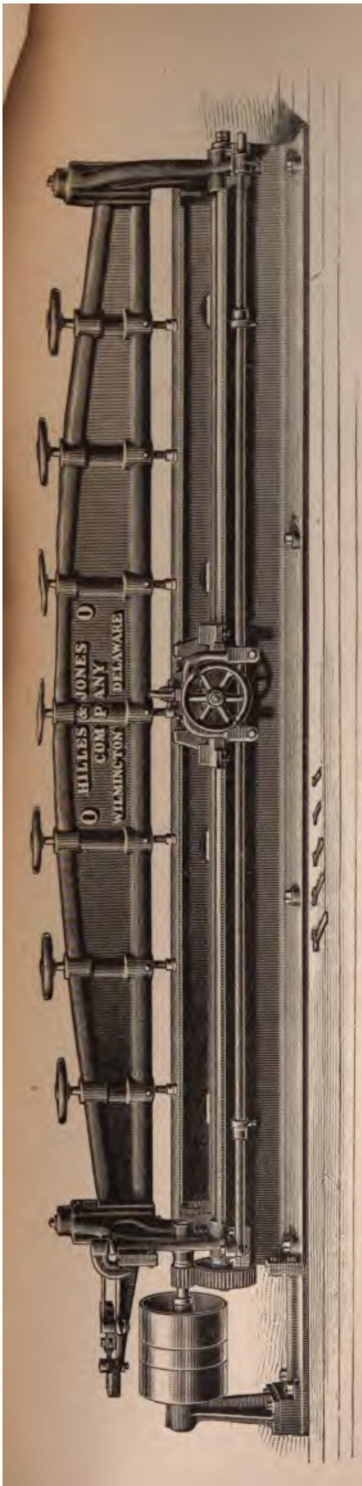

FIG. 106.

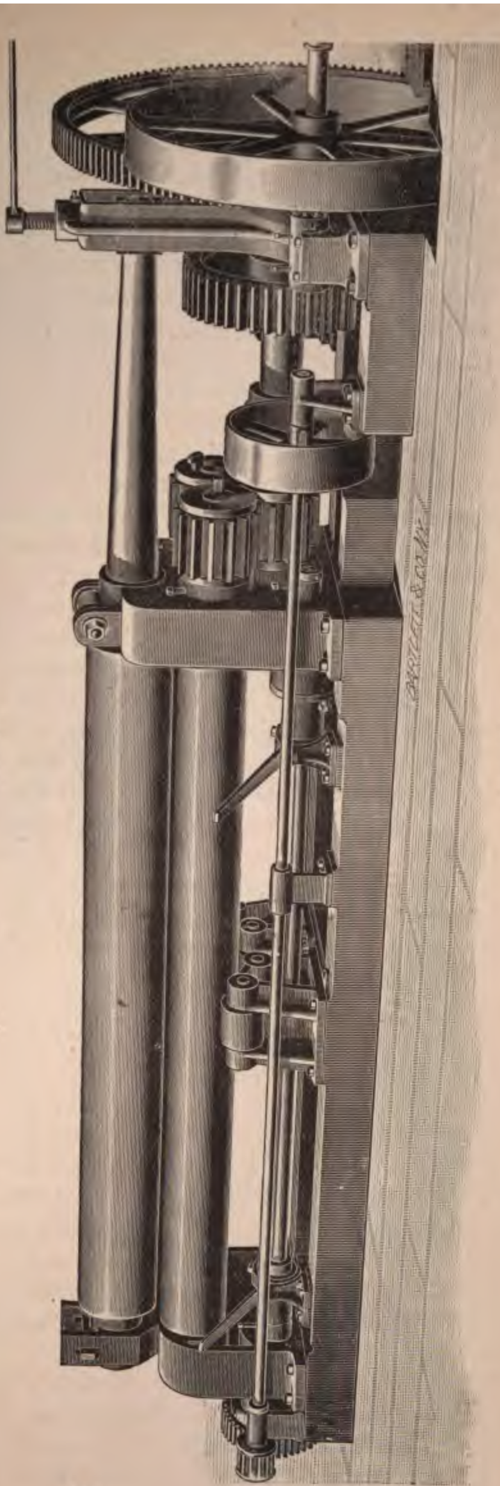

FIG. 107.

pivoting in a movable bearing in the right-hand housing, thus allowing a finished ring of plate to be removed.

Plate IX. gives an illustration of a vertical roll now coming into use.

The movable roll is counterbalanced and is raised clear of the housing when a finished plate is to be removed.

The other plates of the boiler having been treated in the manner described, the different courses are now ready to be bolted together preparatory for drilling and riveting. But few boilers are made to-day with punched holes alone, the common practice being to first punch a hole about $\frac{1}{8}$ inch small, and then to drill out the rest of the hole after the plates are together; this ensures that the holes are in line, and also removes the iron around the edge of the punched hole which was hardened by punching. A punched hole is always larger at the bottom than at the top, on account of the clearance in the die, so when punched holes alone are used, the plates should be put together so that the larger diameters shall be next the rivet heads. The rivet in cooling tends to draw the plates together.

The back head is now put in its course with the flange in, and bolted with six or eight bolts through the punched holes. It is then hoisted up, by means of a crane, to a drill on top of the riveter. Holes are now drilled out at different parts of the circumference, and stay rivets driven.

The bolts are now taken out, as the stay rivets hold the plates together, and the remaining holes drilled out and riveted.

The next course is now put on, bolted in a few places, and then treated in a similar manner.

The reason for putting in a few rivets in different parts and then filling in, rather than taking the holes in order, is that, in the first case, any error in the spacing of the two courses is divided among all the rivets, while in the second case it would all come one way.

The remaining courses are put on in the same way, the front head having the flange facing outward. This enables all the seam riveting to be done by the machine riveter.

As the work progresses, the boiler is hoisted up on end by means of a crane under control of the operator of the riveter.

The greatest width of plate that can be riveted depends upon what is called the gap, or reach, of the riveter, or the height of the post against which the head of a rivet rests while being driven.

Rivets for the riveter are heated in a small forge near by, and are thrown to a man who stands underneath and inside the boiler. This workman picks up the rivets with long tongs, and passes them through the holes from the inside.

The holes for rivets are drilled about $\frac{1}{16}$ inch larger than the rivet. As the rivets are driven under a pressure of from 20 to 70 tons, according to the thickness of plate and size of rivet, they fill the holes and also exert considerable shrink grip to draw the plates together.

The large riveters now in use may be classed as follows: —

(1) Cam and toggle;
(2) Hydraulic;
(3) A combination of (1) and (2);
(4) Steam riveters.

(1) The cam and toggle riveter consists of a toggle closed by a cam driven by a belt and gearing. There is a wedge behind the heading tool adjusted by a screw so as to compensate for laps and different thickness of metal, thus regulating the pressure on the rivet.

(2) The hydraulic riveter, in one form or another, is in most general use to-day. With it a constant pressure can be maintained on rivets in any thickness of plate. Many have a hydraulic closing device which presses the plates together and holds them under pressure while the rivet is being driven, but, as a general rule, workmen do not stop to use it. This type of riveter requires an accumulator or the equivalent of one.

Commonly a large steam pump delivers water into the lower end of a vertical cylinder raising a plunger weighted with pig iron. The supply for the hydraulic cylinder of the riveter is taken from this accumulator at a constant pressure.

(3) The third type of riveter consists of a toggle closed by a cam driven by a belt through a train of gears, and of a hydraulic cylinder on the riveter, and an accumulator located in any convenient place. One end of this toggle presses on the rivet, while the other bears against a plunger fitting in the hydraulic cylinder which is fastened to the frame of the riveter. The back end of this cylinder is connected by a pipe with the accumulator, which is vertical, and has a plunger loaded with suitable weights. Suppose that the vertical cylinder is weighted so that the pressure on the horizontal plunger is 30 tons. Now, as the cam closes the toggle, the back or plunger end will remain still until the pressure on the rivet is 30 tons, then the front end will become fixed, and the back end will push the plunger into the cylinder, driving water back into the accumulator and raising the accumulator weights. The cylinder may be very short, as there is only a slight motion to allow for. There is no loss of water except by leakage. The pressure on the rivet is constant, no matter what is the thickness of the plate, but it can be varied in amount by changing the weights on the accumulator.

(4) The fourth type, the steam riveter, is now not in general use, except in very small machines and in portable riveters.

Figure 108 illustrates a 96″ hydraulic riveting machine. It can be made to exert a pressure on the rivet up to 150 tons, the pressure of course depending upon the weights on the accumulator.

The cylinder and ram are both of steel and have flat upper surfaces, allowing flanged plates of any dimensions to be riveted in any position.

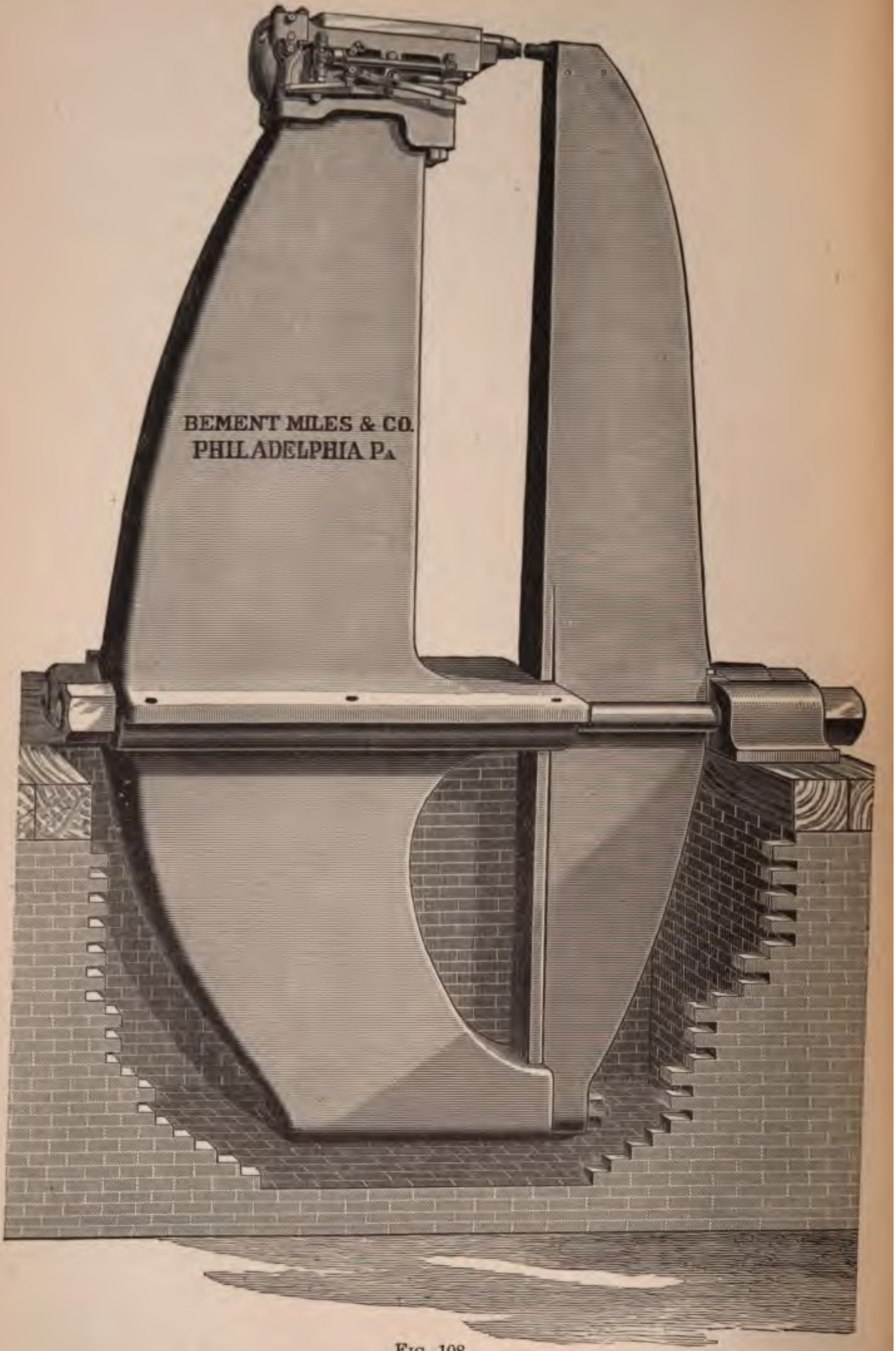

FIG. 108.

The riveting ram being of rectangular section, permits the use of offset dies without fear of the ram rotating. The stroke is adjustable in length and is operated by a single movement of a hand lever. A plate-closing device, operated by a second lever, is sometimes attached.

Figure 109 shows an accumulator with 10″ ram 15′ stroke, such as is used with hydraulic riveters.

The outer casing of boiler plate serves merely to hold pieces of scrap iron used as weights. It is guided by the I beams, as shown. The plunger is fixed to the bottom frame, and fits into a chamber inside of the wrought-iron tank for holding the weights. As water is pumped in, the chamber, tank, and weights lift, the plunger remaining fixed. A packing box on the bottom of the chamber prevents water from leaking out around the plunger.

Water for the riveter is taken through a small hole running down through the centre of the plunger and out at the bottom, as shown. Water coming from the pump is also delivered through the plunger into the chamber. The speed of the pump is controlled automatically by the height of the accumulator.

Plate X. gives an illustration of a 96″ steam riveter with 43″ steam cylinder. The piston rod is made of steel, and is so arranged that it will not turn, so that riveting may be done with offset dies when required. It can be used to drive rivets with pressure or with a blow, as may be preferred. The steam valve is perfectly balanced, so that it works easily. The steam used for driving the rivet also returns the piston before escaping.

A 75 ton portable hydraulic riveter 12″ gap, such as is often used on bridge work and occasionally on boiler work, is shown by Fig. 110.

Steam nozzles, manhole rings, brackets, reinforce plates, and in some cases part of the staying, are often riveted on by the machine. When completed at the riveter, the boiler is lowered to a horizontal position and placed on roller chairs.

The stay rods are now put in, and any small jobs, such as tapping for gauge cocks, feed, and blow-off pipes, are done.

The tubes are now put in and fixed at one end, generally the end hardest to get at, and the other end is cut off to the right length by a tube cutter. The tubes project from the heads a slight distance, so as to allow of beading over. They are expanded so as to be steam tight by means of a tube expander, the ones in most general use being the Prosser and the Dudgeon.

The Prosser expander is made up of a number of segments held in place by an encompassing spring. The outside of these segments is shaped to such forms as it is desired to have the tube set to. The insides of them are made to fit a tapered mandrel which is driven in with a hammer. This tool requires frequent loosening and shifting around on account of the spaces left between the segments.

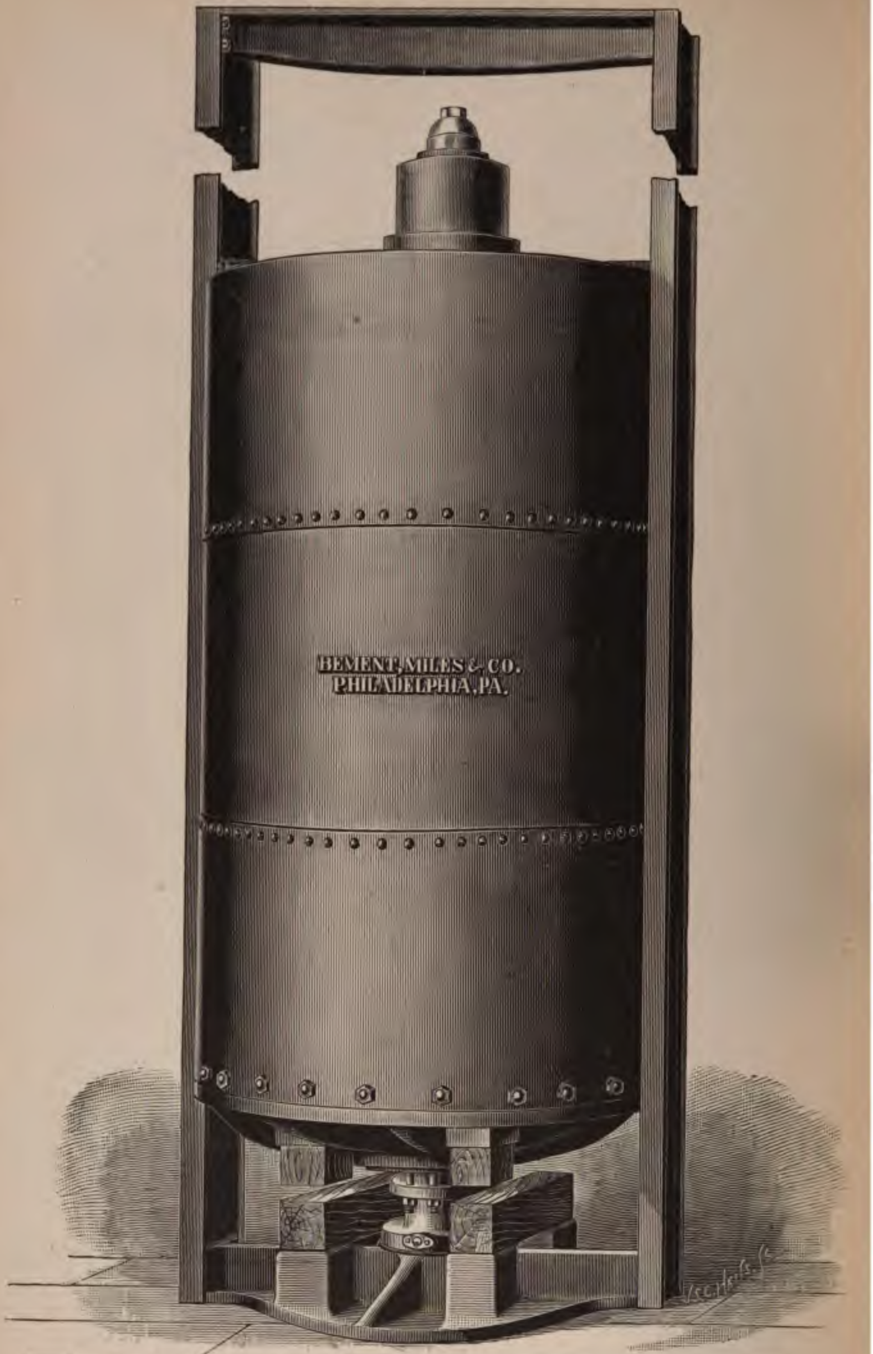
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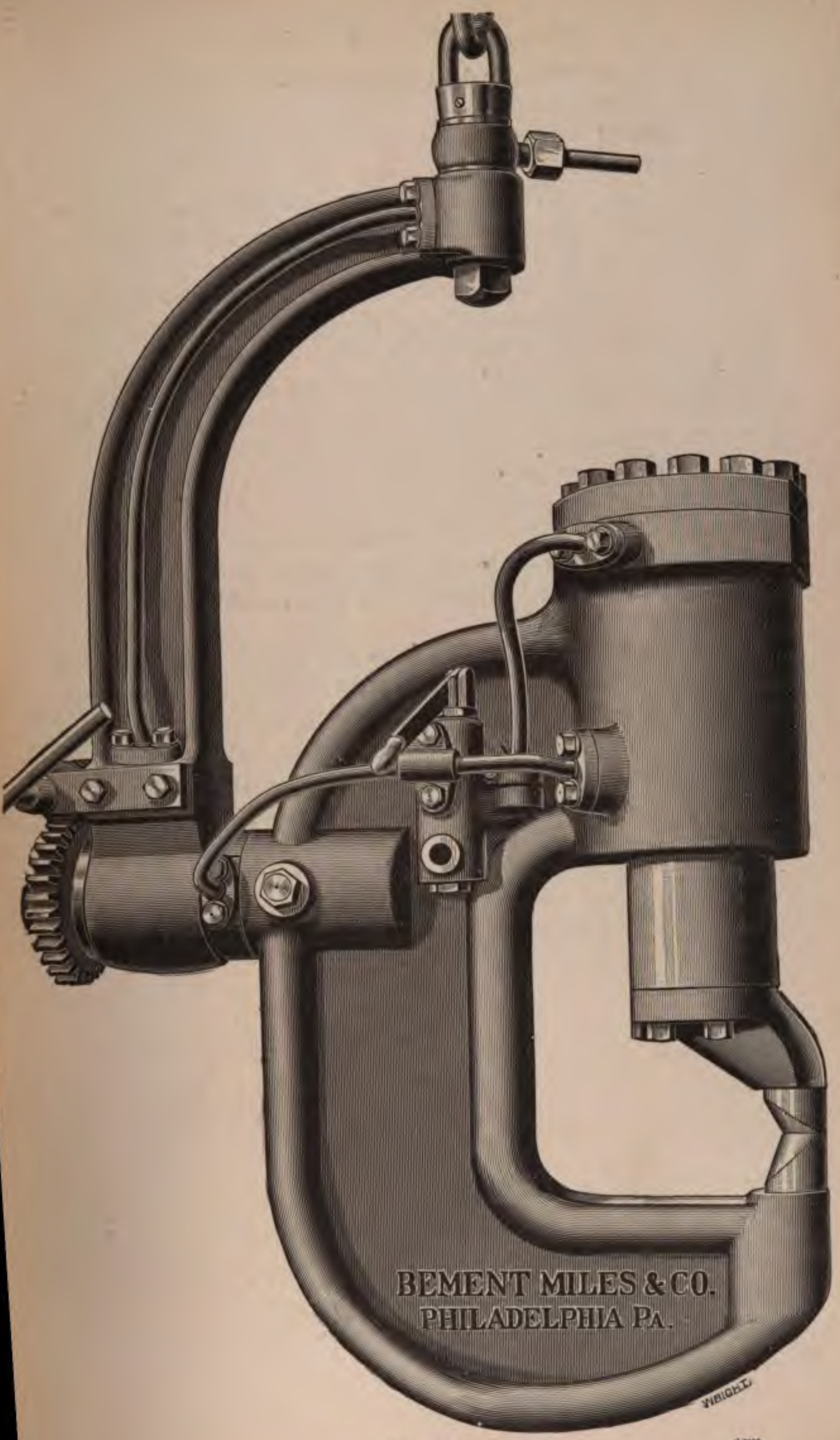
FIG. 109.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PHILADELPHIA PA.
WRIGHT

FIG. 110.

The Dudgeon expander does first-class work. It consists of a series of small rolls of proper shape for forming the tube, forced out by a tapered mandrel which is kept turning while being driven in, thus spreading the tube.

After expanding, the tubes are generally beaded over. The tubes serve to tie the lower part of the heads together, so that it is not necessary to stay them at that point.

The boiler is now ready for caulking. This consists in driving the lower portion of the planed edge of the upper plate against the lower plate, so as to make a water-tight joint. Formerly this work was done with a hammer and caulking tool, but at present it is done by what is called a pneumatic caulker. It is similar in principle to a rock drill. It is connected by a flexible tube with an air compressor, supplying the air needed to run it at a pressure of 60 or 80 pounds.

The whole tool is about $2\frac{1}{2}''$ diameter and 15″ long. It is held by a workman who presses it slowly along the seam, the piston carrying the caulking tool making about 1500 working strokes, $\frac{3}{16}''$ long, per minute. With this tool one man can do as much work as can four working by the old method.

The boiler is now ready for testing and inspection. To do this the openings are closed, and the boiler filled with water, a small air vent being left in the highest point, until all the air is expelled. Water is now pumped in until the pressure is one and a half times the working pressure. The sheets are now examined, to see if there are any flaws or bad leaks. Small leaks are of no consequence, as they soon fill with rust.

If examined and passed by a boiler insurance inspector, he stamps certain letters over the fire door. This inspection should be made before the boiler is painted.

CHAPTER XIII.

BOILER TESTING.

In testing a boiler for its evaporative efficiency, it is necessary to know the following: —

1. Number of pounds of water pumped into the boiler during the test.
2. Number of pounds of coal burned during the test, and the weight of ash left.
3. Temperature of the feed water entering the boiler.
4. Pressure of the steam.
5. Condition of the steam as to moisture.

Water. — The only sure way to measure the water going into the boiler is by direct weighing of the feed water. This can best be done by arranging the suction pipe of the feed pump or injector so that it draws from some sort of a reservoir near the place where the weighing is to be done.

The weighing barrels should rest on scales placed above this reservoir, and should have quick-opening lever valves discharging into it.

Water should be furnished to the weighing barrels either from the city main or by a separate pump. If a pump is used, there should be a by-pass of ¼″ pipe connecting the suction and delivery sides of the pump, so that it may run although the valves leading to the barrels are shut.

At the instant of starting the test, the level of the water in the reservoir should be marked, and at the end of the test should be brought to the same level. A good way to mark this level is to fasten to the reservoir a piece of wire bent in the form of a hook, the point of the hook projecting just above the surface of the water.

At the starting time the level of the water in the boiler should be marked on the gauge glass. This can be done by tying a fine string around the glass, or better, by fastening a wooden scale to one of the gauge rods.

All returns leading to the boiler must be disconnected so that nothing can enter the boiler except through the weighing barrels. As much steam as wanted can be taken from the boiler, but no water should be taken from it.

Coal and Ash. — As it is impossible to estimate exactly the amount of coal on the grate, it is necessary, especially in a test 10 to 12 hours long, to start with a clean grate and ash pit.

A few minutes before starting the test, the fire is drawn and the ash pit cleaned, then, at the instant of starting the test, a known weight of wood is fired, and soon after this, coal previously weighed. Each pound of wood is equivalent to 0.4 of a pound of coal.

All the coal fired during the test is weighed, the record being kept so that the amount fired up to any time can be told approximately.

At the end of the test any coal not fired is weighed back. The fire is drawn and the coal sorted over, the good coal being weighed back; the remainder going in with the ashes which are drawn from the ash pit, and also weighed.

It is sometimes impossible to draw the fires from large boilers. In such cases the condition and thickness of the fire at starting is noted, and at stopping the fire is brought to the same conditions. When fires are not drawn, the test should be at least 24 hours long, preferably longer.

Temperature of Feed Water. — This should be taken close to the check valve at the boiler, and after passing through steam feed-water heaters, if any are used.

While feeding, the temperature should be taken every five minutes. The thermometer should be placed in a thermometer cup, filled with oil, screwed into a fitting on the feed pipe, or into the pipe itself.

Pressure of the Steam. — This should be taken every 15 minutes throughout the test. It should be the same at the end of the test as at the beginning, and should not vary much throughout the test.

The steam gauge should be tested, and corrections made for the error of the gauge and for the water column, if there is any.

The barometer should be read every hour. To reduce the barometer, reading in inches, to pounds, multiply by 0.491.

Priming. — The quality of the steam should be determined at frequent intervals by means of a calorimeter.

Unless the steam is very badly primed, the throttling calorimeter can be used.

Steam for the calorimeter should be taken from the top or the side of the main pipe near the boiler. The nipple entering the main pipe should have a long thread cut on it, so that it will project some distance into the pipe.

Boilers which are worked beyond their capacity are apt to give a large per cent of priming.

Precautions. — To be certain that the brickwork, grate, etc., are hot, and that the boiler has not been drawing heat from them, the fire drawn from the grate at the beginning of the test should be hot enough to generate steam.

The fire drawn at the end of the test should also be able to make steam. Apparently high but erroneous evaporations are sometimes obtained by partially shutting the steam valves on the boiler before the end of the test and allowing the fire to burn down; by so doing, the fire can be burned entirely out, enough heat coming from the walls and grate to keep up the pressure.

When steam is drawn rapidly from a boiler, the water level appears to rise; this has to be looked out for at the beginning and end of the test.

In connection with the tests made by the seniors on the boilers of the Institute samples of flue gas, pyrometer readings and draught pressures are taken both from the stack and from different parts of the setting, and the amount of air entering the ash-pit doors is calculated.

Flue Gases. — The gas samples are drawn from the chimney and setting by means of water pumps. The samples are analyzed in the Orsat apparatus, which gives the number of volumes of CO_2, O, and CO in 100.

Draught pressures are taken by means of the draught gauge shown in Fig. 111. It consists of two pieces of 4″ brass pipes of the same internal diameter, connected at the bottom by a 1″ pipe.

One of these 4″ pipes is sealed at the top, a $\frac{1}{2}''$ pipe with valve serving to connect with the boiler setting.

The other pipe, which is open, has a hook gauge reading to .001 of an inch running down into it. The two pipes are filled about half full of water; the $\frac{1}{2}''$ pipe being open to the air, the level will stand the same in each. The point of the hook gauge is brought to the surface of the water and a reading taken. The $\frac{1}{2}''$ pipe is now connected with the setting. The level in the open leg will drop. Twice the difference between this reading and the former reading will give in inches of water the difference of pressure.

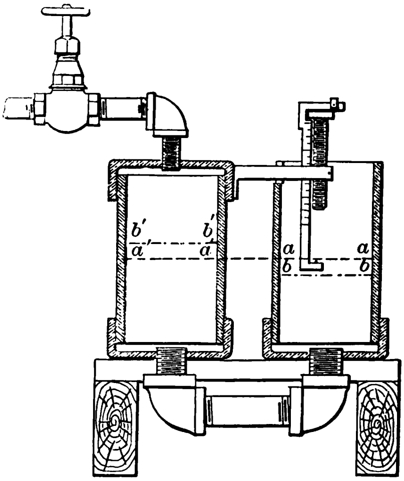


Fig. 111.

The pipe leading through the setting should be at right angles to the direction of the current.

Pyrometer. — The pyrometer used in the tests made at the Institute consists of two wires, one of platinum and the other of platinum alloyed with 10 per cent of rodium, joined at one end, all but about

4″ of the wires being incased in fire clay inside of an iron pipe about 4 feet long. From these wires two other wires lead to a galvanometer placed in another room.

A ray of light is thrown from a lamp onto a mirror on the galvanometer needle and reflected back onto a scale graduated arbitrarily.

The scale is moved in or out until the ray of light is on the zero of the scale when the wires are in the air. When the wires at the free end of the iron pipe are heated, a thermo-electric couple is formed which causes the galvanometer needle to be deflected and the ray of light to travel along the scale.

To find the temperature corresponding to any reading, a curve is plotted with scale readings as abscissæ and known temperatures corresponding to different scale readings as ordinates.

To determine the scale reading for known temperatures, the wires forming the hot junction are immersed in melted sulphur, naphthaline, etc., and the scale readings taken. To the temperature reading as taken from the curve at the abscissæ corresponding to the scale reading, the temperature of the room has to be added.

Air Supply. — To determine the amount of air entering the ash-pit doors, a casing of galvanized iron is fitted into the door openings, the opening into this casing being a circular pipe about three feet long and of an area of cross section as great as that of the two doors. An anemometer is placed in this pipe, and its readings show the velocity which, multiplied by the area of the pipe and by the weight of a cubic foot of air at the temperature at the mouth of the pipe, gives the weight of air.

Sample Test. — The data and results of a 128-hour test are given on page 161.

BOILER TEST.

DATE, *Dec. 28, '91–Jan. 2, '92.*

Duration of Test . . . . *128* hours.

Average pressure of air . . . . *14.85* lbs. per sq. in.

" gauge pressure . . . . *109.9* " " " "

" temperature of feed water . . . . *50.51* C. *122.92* F.

	Boiler No. *1*.	Boiler No.
1. Description of Boilers: *Babcock & Wilcox, No. 1* . *108 tubes 4″ dia., 17′-8″ long; outside area, 1997.3* *12 " 4″ dia., 4′-6″ " " " 56.5* *2 drums 3″ dia., 17′; one-half of shell, 160.2*		
2. Grate surface, No. *1*, *84* in. by *88* in. No. ___, ___ in. by ___ in. Area, ft.	*51.3*	
3. Water-heating surface, feet	*2214*	
4. Ratio of water-heating surface to grate surface . .	*43.16*	
5. Lbs. coal fired, including coal equivalent of wood .	*64639*	
6. Unburned fuel	[1]	
7. Coal burned, including coal equivalent of wood . .	*64639*	
8. Average coal burned for *15* minutes	*126.2*	
9. Total refuse from coal	*8669.*	
10. Total combustible	*55970*	
11. Average combustible for *15* minutes	*109.3*	
12. Average lbs. of air for *15* minutes	*1681.*	
13. Air per lb. of coal	*13.32*	
14. Air per lb. of combustible	*15.38*	
15. Quality of steam, saturated steam taken as unity .	*.983*	
16. Total water pumped into boiler and apparently evaporated	*548794*	
17. Water apparently evaporated per lb. of coal burned	*8.49*	
18. Water actually evaporated, corrected for quality of steam	*539465*	
19. Equivalent water evaporated into dry steam, from and at 212° F.	*614277*	
20. Equivalent water evaporated into dry steam, from and at 212° F., per lb. of coal burned	*9.50*	
21. Equivalent water evaporated into dry steam, from and at 212° F., per pound of combustible . .	*10.98*	
22. Coal burned per sq. foot of grate surface per hour .	*9.84*	
23. Water evaporated, from and at 212° F., per sq. foot of heating surface per hour	*2.167*	

---

[1] Fires not drawn.

CHAPTER XIV.

BOILER DESIGN.

In the design of a boiler the most important considerations appear to be to provide —

1°. Safety, or strength.

2°. Durability, or strength for the future.

3°. Convenience in cleaning and inspection.

4°. Capacity to do the required work. This is placed last as, relatively, it requires less attention than the others.

All of the above features have been considered in what precedes. In many localities certain laws or rules are in force which, to a fixed extent, govern the general allowable working stresses in various parts of a boiler, and these must be given due consideration. Especially in marine work is it necessary to conform to the inspection rules of the insurance companies, as well as those of the boards of inspection appointed by law. For example, the United States Inspectors' Rules limit the working stress on wrought iron welded stays to 6000 pounds per square inch. The maximum working stress on stay rivets is limited in the same way.

Type of Boiler. — The kind of a boiler to be selected for any particular work or locality depends on many things, and it is seldom advisable to vary much from the existing common practice. Thus, when the feed water obtainable is likely to deposit much scale, or is dirty, the externally fired, horizontal, multitubular boiler would not be desirable; with good, clean, soft, river water, however, this boiler gives very good results. When muddy water, or that depositing much scale, must be used, the boiler should be so constructed as to be easily and conveniently cleaned.

The working pressure to be used also influences the selection. For externally fired boilers it is not advisable to have the thickness of the shell exceed $\frac{1}{2}''$, and better $\frac{3}{8}''$, on account of trouble from overheating. Thus, the diameters allowable for high pressures become small, and some type of sectional boilers is chosen. Quite large externally fired boilers are now constructed for high pressures in which a very efficient longitudinal joint allows thin plate to be used.

The cost of fuel, other things being the same, makes boilers of higher evaporative efficiency desirable, the greater such cost becomes.

For cheap fuel and bad water the plain cylindrical boiler has given good results.

The room available often makes a choice impossible, the restrictions being such as to positively fix the type. Such is the case in locomotives. With small ground room and ample height some type of vertical boiler is made necessary to obtain the power required. In marine practice the type is usually governed by the room available.

The type of boiler having been chosen, it then remains to so proportion its parts as to satisfy the four conditions mentioned, and also to satisfy the inspection rules.

For all types, the amount of water to be made into steam being known, we must provide—

1°. Sufficient grate area to burn the fuel required under the available draught.

2°. Suitable combustion space and flue area to carry off the products of combustion.

3°. Sufficient heating surface to absorb the heat of combustion.

4°. Proper water space, so that an irregular demand for steam may not cause too great a fluctuation in the water level.

5°. Suitable steam space, so that the steam required at intervals will not cause too great a fluctuation of pressure.

For most types the relations between the above-mentioned quantities are known within narrow limits, each type having had its dimensions varied until good results were obtained. These relations may thus consistently be used in designing. The method of procedure can best be shown by means of an example.

Suppose we wish to design a 70 H.P. (A.S.M.E. standard, page 80) horizontal, multitubular boiler to carry a working pressure of 150 pounds per square inch, and to be subjected to a test pressure of 225 pounds per square inch. The fuel is to be anthracite coal; the length of the tubes is to be 15 feet, which is not to exceed 60 diameters; and the heating surface must be at least 37 times the grate surface.

Let G = area of grate in square feet;
C = area of smoke passage through tubes in square feet;
H = heating surface in square feet;
S = steam space in cubic feet;
W = water space in cubic feet.

The water to be evaporated, from and at 212° F., is

$$34.5 \times 70 = 2415 \text{ pounds.}$$

For this boiler we may safely assume that one pound of anthracite coal will evaporate 9 pounds of water from and at 212° F., and at the same time allow some margin for economy. Also that, with reasonable draught, 12 pounds of coal can be burned per square foot of grate, which leaves some margin for an increased capacity.

The weight of coal burned per hour will be

$$\frac{2415}{12} = 201.25 \text{ pounds.}$$

The square feet of grate area will be

$$G = \frac{2415}{9 \times 12} = 22.36+.$$

In this case the tubes must have an outside diameter of at least 3 inches; and for a standard charcoal iron boiler tube we have its internal circumference 8.74″, its internal and external transverse areas 6.08 and 7.07 square inches respectively.

Now assume (page 81) that $C:G = 1:8$; then the number of 3″ tubes required to give sufficient draught area is

$$\frac{22.36 \times 144}{8 \times 6.08} = 66+.$$

Steam Space. — This may vary widely, but the rules given on page 79 will give us a means of finding values in cubic feet per H.P. suitable for use here.

1°. Let us assume our engine to use 30 pounds of steam at 70 pounds pressure per H.P. per hour, and that the steam space shall be sufficient to supply the engine for 20 seconds.

The absolute pressure is 85 pounds nearly, and the specific volume of steam at this pressure is 5.125.

The steam space required per H.P. is

$$\frac{30 \times 5.125 \times 20}{60 \times 60} = 0.854 \text{ cubic feet.}$$

2°. Assume the steam to be used in a triple-expansion engine at 150 pounds pressure, 165 pounds absolute, and take the engine as making 60 strokes per minute, and cutting off at one-third stroke.

If we take the specific volume of the steam as 2.75, and assume the steam space as 60 times that of the cylinder, we shall have the space required per H.P.

$$\frac{13 \times 2.75 \times 3 \times 60}{60 \times 2 \times 60} = 0.89+ \text{ cubic feet.}$$

We will assume 0.85 cubic feet per H.P.

Diameter of Boiler. — The total volume of our boiler is then in cubic feet.

$$\begin{aligned} \text{Steam space} &= S = 0.85 \times 70 &&= 59.5 \\ \text{Water space} &= W = 2\,S \text{ (page 79)} &&= 119. \\ \text{Tube space} &= \frac{66 \times 7.07 \times 15}{144} &&= \underline{48.6} \\ & && \ 227.1 \end{aligned}$$

The area of the end of our boiler is

$$\frac{227.1 \times 144}{15} = 2180+ \text{ square inches.}$$

Or the diameter is 52.7″—.

We will try a diameter of 54″, which will make allowance for the space occupied by the staying.

Heating Surface. — If we allow one-half of the shell and the total inside surfaces of tubes as heating surface, we have

$$\begin{aligned} \text{Heating surface of shell} &= \frac{169.6 \times 15}{2 \times 12} = 105.9 \text{ square feet.} \\ \text{“ “ “ tubes} &= \frac{8.74 \times 66 \times 15}{12} = \underline{721.} \quad \text{“ “} \\ \text{Total} &= H = 826.9 \quad \text{“ “} \end{aligned}$$

$$\frac{H}{G} = \frac{826.9}{22.36} = 37, \text{ very nearly.}$$

The heating surface of the back head will increase this ratio slightly. In arranging the tubes, it will be best if possible to put in more than 66.

When the ratio of H to G comes low, the number of tubes must be increased and a new boiler diameter determined. The draught area will, of necessity, also be increased.

We will make the boiler 54″ diameter at the outside of the inside shell, with 66, or more, 3″ tubes and a grate 54″ wide and 60″ long.

Water Level. — To have $W = 2\,S$, we should divide the transverse sectional area of the boiler into two parts in the ratio of 1 to 2. One method of easily doing this with the aid of a table of areas of segments of a circle is as follows —

The transverse area is that of a 54″ circle, minus the combined areas of the tubes, or

$$2290 - 66 \times 7.07 = 1823.4 \text{ square inches.}$$

One third of this, or 607.8 square inches, is the area of the segment occupied by the steam. From the table of segments we find the height of this segment = 16.8″.

To give the required steam space $S = 59.5$, not allowing for space taken by the stay rods, the segmental area will be

$$\frac{59.5 \times 144}{15} = 571.2.$$

The corresponding height is 16″.

If the tubes can be arranged suitably, it will be best not to have the mean water level more than $27 - 16.5 = 10.5''$ above the centre line of boiler.

Tube Sheet. — The tubes should be arranged in horizontal and vertical rows if possible. They may be placed at the intersections of lines making an angle of 30° with the centre line (Fig. 56), if this is necessary to get in the given number. The distance between tubes should not be less than $\frac{3}{4}''$, and had better be 1″. It is advisable, for the sake of good circulation, to make the horizontal greater than the vertical spacing, also to secure a central circulating space, as shown in Plate I., when possible.

The common range of water in a boiler of this size is 6″, 3″ above and 3″ below the mean water level. The tops of the tubes are usually set 3″ below "low water," which in this case would be $10.5 - 6 = 4.5''$ above the centre line of the boiler. The tubes should not be placed nearer than 3″ to the shell, and should be from 4″ to 6″ above the bottom of the boiler. The hand hole should be located as low as possible, the flat surface for a gasket being at least $\frac{3}{4}''$ wide; and no tube should be nearer than 1″ to its edge. A tube sheet fulfilling these requirements, except that the top row of tubes is placed $\frac{1}{2}''$ higher, is shown in Plate I. Were the central space omitted, it would be possible to get in more than 68 tubes 3″ diameter.

The tube sheet is usually made from $\frac{1}{16}''$ to $\frac{1}{8}''$ thicker than the boiler shell. The internal radius of the flange should not be less than $\frac{1}{2}''$; for plates $\frac{1}{2}''$ and under the outside radius is often made 1″.

It often happens that the required number of tubes cannot be put in without raising the water level, as is the case in Plate I. If the reduction in the steam space thus caused is not permissible, the diameter of the boiler must be increased. It should be borne in mind that, for a given draught area, the smaller the tube diameter, the greater will be the tube-heating surface; since the former varies with the square of the diameter, and the latter as the diameter. Increase of length also adds very materially to the heating surface. A trial by drawing is the only way to decide on the number and arrangement of tubes allowable. Such drawing, especially if on a small scale, should be very carefully made to be of any value.

If we waive the $\frac{1}{2}''$ increase in height of our calculated water level, the boiler shown in Plate I. fulfils our requirements. The number of tubes is 68, which will secure a value of $\frac{H}{G}$ greater than 37. While Plate I. was not intended for a solution of this problem, and while some changes might be made for the better, yet it represents the case sufficiently well to be used as a basis for the remaining work of design. The dimensions given on the plate are correct, but the scale is not, in some places.

Factor of Safety. — This is usually made from four to five, and in places where the stresses cannot be exactly determined it is often much greater. In the case of stay rods and stay rivets, the allowable stresses are limited by the inspection rules, and sometimes by law. In deciding on a factor, corrosion should be allowed for, a definite amount being assumed to take place. In the case of a round rod 1″ diameter, we would have only a $\frac{7}{8}''$ rod after $\frac{1}{16}''$ corrosion, and the strengths before and after would vary as the areas 0.7854 and 0.6013. In the case of a plate the effect would not be so marked, especially in the joint, the weakest part, where the plate between rivets is protected by the rivet heads; yet the effect on a thin plate is much greater than on a thick one. Many forms of joints have cover plates that protect the weakest sections from corrosion. With reasonable care no corrosion should take place on the outside of a boiler properly set.

We will suppose a maximum corrosion of $\frac{1}{16}''$ to take place on the inside of the boiler, and so proportion the parts that there will be at least a factor of safety of *four* after corrosion has taken place. The proportions for the joints will, however, be determined from the solid plate and will then be tested for corrosion. When there is doubt about the exact stresses, a higher factor will be used. In all cases, the sizes should be such that there is no danger of exceeding the limit of elasticity of any part under the test pressure of 225 pounds per square inch.

Thickness of Shell. — For a trial thickness it is usual to allow a factor of safety of six on the solid plate. This thickness may later be reduced if the efficiency of the joint chosen is high enough to warrant it, due allowance having been made for corrosion. In deciding on the thickness, the joints on the manhole ring and steam outlets should be considered as well as the longitudinal joints.

We will assume the tensile strength of the steel plate for this boiler as 55,000 pounds per square inch. For a factor of six, the thickness of plate will be (page 103)

$$t = \frac{pr}{f} = \frac{150 \times 27 \times 6}{55000} = 0.442 = \tfrac{7}{16}'', \text{ nearly.}$$

We will use $\frac{7}{16}''$ steel plate and make the tube sheets $\frac{1}{2}''$ thick.

Longitudinal Joint. — Let p = pitch on the outside row of rivets; d = diameter of rivet, and a = area of rivet after driving; t = thickness of plate; f_s = shearing strength of rivet; f_t = tearing strength of plate. All dimensions are in inches, and stresses are in pounds per square inch. To be on the safe side, no allowance will be made for the excess tenacity due to the grooved specimen, which will tend to offset loss of strength due to corrosion. In deciding on the proper pitch, it is customary to assume that the joint will fail equally well by shearing the rivets or by tearing the plate, the weakest section being chosen. At the same time care is taken that the bearing pressure on the rivet is not too great. An excessive bearing pressure tends to decrease the strengths of both rivet and plate.

In the joint shown, commonly known as a "welt-plate" joint, lines drawn through the rivets in the outer rows give a repeating section. Since the outer plate is not as well situated as the inner, so far as shearing on rivets is concerned, it will be necessary to base our calculation upon the former. The joint may fail in one of three ways:—

1°. The plate may tear on the outer row of rivets, the resistance being $(p - d)tf_t$.

2°. The plate may tear on the inner line of rivets and shear one rivet on the outer line, the resistance being $(p - 2d)tf_t + af_s$.

3°. Three rivets may shear with a resistance of $3af_s$.

Equating 1° and 2°, a diameter of rivet may be found which will give equal tearing strength for both sections. This diameter need not, however, be used, provided the strength on the inner row is as great or greater than that of the outer row. From 1° and 2°, replacing a by its value in terms of d, we have

$$d \geq \frac{tf_t}{0.7854 f_s}.$$

For iron rivets and steel plate let us take $f_t = 55000$, $f_s = 38000$, and $t = \frac{7}{16}$; then $d \geq 0.81''$. For steel rivets, d would come even smaller, since f_s is larger.

Equating 1° and 3°, we have

$$(p - d)tf_t = 3af_s,$$

$$p = \frac{3af_s}{tf_t} + d.$$

The value a has been used in place of $0.7854d^2$, because it is much easier to look it up in a table of areas than to make the calculation.

We will use steel rivets, except in the staying, where iron will be used.

For steel rivets

$$d \geq \frac{tf_t}{0.7854 f_s} = \frac{7 \times 55000}{16 \times 0.7854 \times 50000} = 0.61''+.$$

We will use $\frac{13}{16}''$ driven rivets; then

$$p = \frac{3 \times 0.518 \times 50000 \times 16}{7 \times 55000} + \frac{13}{16} = 4.04''+.$$

The pitch on the inner row is 2.02″.

With this pitch the maximum compression on the bearing surface will be, for the bearing area $3\,dt$,

$$\frac{(p-d)tf_t}{3\,dt} = \frac{(p-d)f_t}{3\,d} = \frac{3.23 \times 55000}{3 \times 0.812} = 72{,}926,$$

which is allowable.

Ring Seam. — Since the ring seams come in the fire, and since the stress on them cannot exceed one-half that on the longitudinal seams, it is customary to make them single riveted. The pitch used usually lies between that due to a double-riveted lap joint and that of a single-riveted lap joint.

For equal shearing and tearing strengths, the pitch for double riveting would be, for $\frac{13}{16}''$ rivets,

$$\begin{aligned} p &= \frac{2\,af_s}{tf_t} + d \\ &= \frac{2 \times 0.518 \times 50000 \times 16}{7 \times 55000} + \frac{13}{16} \\ &= 2.15 + 0.81 = 2.96''. \end{aligned}$$

For single riveting,

$$p = \frac{2.15}{2} + 0.81 = 1.88''.$$

A mean of the above would be 2.42″. We will allow 74 rivets in the ring seam, thus making the pitch on a 54″ circle = 2.29″+.

Lap. — The lap, or distance from the centre of the rivet to the edge of the plate, is usually taken 1.5 *d*. It is approximately determined by considering the part of the plate in front of the rivet as a beam loaded in the middle, with a load equal to the shearing strength of the rivet, and fixed at the ends. In case of double shear, the load is twice as great. For single shear, we have in this case, when *h* is taken as the distance from the edge of the rivet to the edge of the plate,

$$\frac{af_sd}{8} = f_t\frac{th^2}{6};$$

$$h = \sqrt{\frac{3\,af_sd}{4f_tt}} = \sqrt{\frac{3 \times 0.518 \times 50000 \times 13 \times 16}{16 \times 4 \times 55000 \times 7}} = 0.81.$$

The lap is $0.81 + 0.406 = 1.22-$. The rule first stated gives the same result, and in most cases is outside this value, especially for iron rivets.

The cover plate, or welt plate, on the longitudinal joint is usually made $\frac{1}{16}''$ thinner than the plate. The lap in such case would be, using the same beam theory and substituting $\frac{3}{8}''$ for t, $0.87 + 0.406 = 1.28''-$. The results obtained by this rule, while not theoretically correct, have been found to be sufficient in practice. We will make the lap 1.25″ in all cases, including the inner part of the longitudinal joint, which is never stressed enough to bring the double shear on the rivet into play.

Area of Uptake. — This is usually made one-eighth of the grate area, which gives

$$\frac{54 \times 60}{8} = 405 \text{ square inches.}$$

Make the opening 12″ × 34″, the latter dimension being taken on a chord.

Length of Sections. — The length of the sections depends on the "reach" of the riveting machine and the size of plates obtainable. For the sake of uniformity it is well to have the plates of all sections the same in width, provided it is possible in so doing to keep the ring seam out of the most intense heat near the bridge wall and to satisfactorily space the joint. We will assume the reach of the riveting machine six feet, and also that plates can be obtained long enough to pass entirely round this boiler, thus making each section of one piece.

We will take 1″ as the outside radius of our tube sheet, which, with $1\frac{1}{4}''$ lap, gives us $2\frac{1}{4}''$ as the distance to the rivet line from the outside face. Also allow the plate at the front end to project $\frac{1}{8}''$ in order to protect the caulking at the uptake opening.

This gives for the width of each plate, allowing for five $1\frac{1}{4}''$ laps, and for three sections:

$$\frac{180 + 1\frac{3}{4} + 13\frac{3}{8} + (5 \times 1\frac{1}{4}) - 2\frac{1}{4}}{3} = 66\frac{3}{8}''.$$

The distance between rivet lines is then $66\frac{3}{8} - 2\frac{1}{2} = 63\frac{7}{8}''$. When a remainder exists in dividing, it is always possible to change the width of the uptake enough to bring the result even.

With this width of plate, the ring seam on the left-hand section comes about 14″ in front of the bridge wall. The joints are now carefully located and dimensioned, as shown, to the rivet centre lines, from which all laying out is done.

The middle section is commonly placed inside, which favors the joint over the fire and brings the manhole, usually located near the middle of the boiler, on the smaller diameter.

It now remains to space the rivets on the longitudinal joints. The joints for the middle and right-hand sections may be spaced the same. The end rivets on the middle row are placed as near as possible to the ring seams, $\frac{1}{8}''$ to $\frac{1}{4}''$ being allowed between the edge of the plate and the edge of the rivet head. Allowing a rivet head $2d$ diameter and $\frac{3}{16}''$ space, we obtain $2\frac{1}{4}''$. The problem then is to arrange our rivets in the resulting distances $59\frac{5}{8}$ and $47\frac{1}{4}$, and leave the pitch as near as possible to the one found by calculation in both places. If in the longer joint we use 30 spaces, and in the shorter 24 spaces, we will obtain 1.98″ and 1.97″, respectively, in place of the calculated pitch 2.02″. If a closer approach to the calculated pitch is wanted, it can be obtained only by a change in the breadths of plates, which would be desirable especially in the more complicated joints, where a symmetrical arrangement and exact spacing is more important than having the plates the same in width. The spacing determined above will be used, giving the arrangement shown on Plate I., which brings the pitch for both inner and outer rows nearly uniform. The rivets in the outer row are always arranged opposite either a rivet or the centre of a space of the inner row, and are located far enough away to allow easy bending of the welt plate. The distance is at least twice the lap, and in this case had better be a little more.

The welt plate is discontinued at the front tube sheet, and the joint in the smoke box is single riveted, as shown. The middle row of rivets is always placed in line with a rivet in the ring seam. In order that this joint may take the stress upon it without distortion, the welt and boiler plate should be very carefully bent to bring the stresses central. The method of scarfing the plates is clearly shown.

Efficiency. — The theoretical efficiency of this joint as arranged is

$$\frac{p-d}{p} = \frac{2 \times 1.97 - 0.812}{2 \times 1.97} = 0.79+.$$

The factor of safety is, — there being no corrosion on account of the protection given by the welt plate, — with a working pressure $P = 150$ pounds,

$$\frac{(p-d)tf_t}{pP27} = \frac{3.128 \times 7 \times 55000}{16 \times 3.94 \times 150 \times 27} = 4.72.$$

With this factor there is no danger of stressing the joint beyond the elastic limit at the test pressure.

A butt joint with an extended cover plate on the inside, similar to Fig. 65, but with only two rows of rivets passing through both cover plates, would be better than the one used, and would be easier to construct. The longitudinal joints are arranged to alternate on each side of the boiler and in the steam space when possible, as shown on the end and sectional views.

The joints for the manhole ring and the steam outlets, on account of their rounded shape and the stiffness of the boiler plate, are better situated to take the hoop tension upon them than are the longitudinal joints. For pressures of 150 pounds and over, it is best to use a double row of staggered rivets, as shown, and, on account of the curvature, the pitch of the outer row may be made somewhat greater than that calculated for a double-riveted lap joint, which tends to increase the efficiency. The perpendicular distance between the rows is made such that the plate section cut in passing from an outside rivet to an inside rivet and back to the next outside rivet diagonally, is at least 30 per cent greater than the section cut in passing directly from one outside rivet to the next.

For a double-riveted lap joint, the pitch was found, page 169, to be 2.96″. A distance somewhat greater than this, 3⅛″ to 3¼″, should be so spaced on the developed outside rivet line as to secure an even number of rivets. The arrangement should in both cases be the same as shown on Plate I., but the number of rivets will be found somewhat less than that on the plate, which is arranged for iron rivets.

After locating the rivets on the development, a central strip of plate — limited by parallel lines passing through the centres of two rivets or the spaces between rivets, and about one-half the distance between the outer rows of rivets longitudinally in width — is tested, and its factor of safety found by a method similar to that used in the longitudinal joint. From the chosen width, the hoop tension is determined; the number of rivets and the tearing area are found by inspection.

Manhole. — The manhole ring should be strong enough and stiff enough to take care of the hoop tension belonging to the length of plate cut out longitudinally. It is made as small as is consistent with the possibility of entering the boiler, and its longer dimension should always be placed across the boiler. A lip at least one inch wide is provided on the inner edge for seating the cover plate. The bearing surfaces of cover and ring should be finished, yet this is not always done; in either case a gasket is used to insure tightness, the cover being held in place by a bolt and yoke. For boilers carrying over 100 pounds pressure, the rings are made of good gun iron or, better, of steel castings.

Inasmuch as the determination of the stresses in the ring becomes exceedingly complicated, especially when eccentric loading is taken into account, and at best can only be approximate, it will not be taken up here. The sizes successfully used in practice furnish the best data for proportions. A rule, often applied by the boiler makers, for the styles of ring now in use, is to put material enough in the ring section, not counting the lip, to replace the plate cut out, section for section, due allowance being made for the difference in tensile strengths of the materials used and blow holes likely to occur in castings, also for the areas removed by the rivets in the ring. The size of the ring on the steam outlet may be found in the same manner, but here the cylindrical portion adds much strength and stiffness. In either case, the rings must be wide enough to take the rivets and at the same time allow them to be arranged centrally. Rings of gun iron are seldom made less than 1½" thick for the manhole and 1¼" for the steam outlets. Steel rings may be thinner. Wrought iron rings are now made for both the internal and external forms.

The thickness of the manhole cover is determined by considering it as a flat plate subjected to a uniformly distributed load per square inch equal to the steam pressure, also to a concentrated load in the centre brought about by the spring of the yoke due to tightening the cover. The cover is usually made of cast iron, with a raised lip to keep it in place and handles for convenience in removing. For higher pressures steel will soon be used.

The hand-hole plates are calculated and arranged in the same manner as the manhole cover. The joint is made by a rubber gasket at least ¾" wide. For thin plates the yoke is cast upon an elliptical ring passing around the opening in the tube sheet, and thus furnishing a good support. For so thick a plate the yoke shown is sufficient; it is, however, drawn higher than need be; the distance from its under side to the plate need not exceed 1½". The nut on the hand-hole bolt at the back end of the boiler is sometimes replaced by a taper key, which is more easily removed than a nut which has been intensely heated; in fact, these nuts can seldom be removed without twisting off the bolt. The hand holes are always located as low as possible.

Staying. — To be on the safe side, the flat heads of this form of boiler, and in fact any heads that require staying, are not supposed to have any inherent stiffness, except between the rivets or stay rods used as supports.

In proceeding to stay a flat head, a general method would be as follows: —

1°. Knowing the thickness of the plate, determine the maximum distance allowable between supports without unduly stressing or bending the plate. For a flat plate of thickness t, supported uniformly by stays arranged in lines distant a from each other, as in the side of the fire box, Fig. 56, and subjected to a pressure of p pounds per square inch, the greatest stress is

$$f = \frac{2}{9}\frac{a^2}{t^2}p.$$

To prevent excessive deflection in the plates, we will use a factor of 10 on the maximum allowable fibre stress, thus giving a working stress $f = 5500$ pounds per square inch. Substituting in the transposed expression the known values of f, $t = \frac{1}{2}''$, and p, we find

$$a_1 = \sqrt{\frac{9ft^2}{2p}} = \sqrt{\frac{9 \times 5500}{2 \times 4 \times 150}} = 6.42''.$$

That is, we may locate supports 6.42" on centres without overstressing the plate or causing it to spring excessively.

2°. Select a diameter for the stay rivet and see how great an area can be supported by it. We will in this case assume $\frac{13}{16}''$ as the driven diameter of the iron stay rivet, and will allow a working stress of 6000 pounds per square inch. The area of our rivet is 0.518 square inch, and the side of the square that can be supported by one rivet is

$$a_2 = \sqrt{\frac{6000 \times 0.518}{150}} = 4.55''.$$

Thus $\frac{13}{16}''$ rivets cannot be spaced over 4.55" on centres without overstressing them.

Rivets under $\frac{3}{4}''$ diameter are seldom used for staying; they are under initial stress due to shrinkage, and on that account so low a working stress is selected.

3°. See what influence the style of staying will have on the pitch.

In the case of crowfoot staying, Fig. 50, it is not advisable to make the stay rods, which are attached to the shell, of too large a diameter, since they bring too great a local stress upon the plate. Shell stays are not commonly made over $1\frac{1}{4}''$ diameter, $1\frac{1}{8}''$ and 1" being common sizes.

The maximum number of rivets allotted to one stay is four. If we assume a $1\frac{1}{4}''$ rod, 1.227 square inches area, allow it to be inclined 20° to the shell, and limit the stress along it to 7500 pounds per square inch, the side of the square that can be stayed by each one of its four rivets, or what amounts to the same thing, the distance on centres of its rivets, will be

$$a_3 = \frac{1}{2}\sqrt{\frac{7500 \times 1.227 \cos 20^\circ}{150}} = 3.8''-.$$

Thus, for crowfoot staying, we would be limited to 3.8" spacing for rivets, the smallest of the three spacings.

In a form of staying such as shown in Fig. 48, the larger of the values a_1 or a_2 would be used.

With girder stays, Plate I., the pitch vertically, at least, is limited in part by the size of channel, angle, or tee irons used. In channel iron, the spacing is limited by the width of the channel. With angle irons, the rivet lines should not be far enough apart to cause bending of the tube sheet at the point where the through rods give support, thus causing leakage. That is, assuming the channel in Plate I. replaced by two angle irons, the vertical spacing $3\frac{1}{2}''$ should not be too great. Tee irons, Fig. 48, top girder, would become too heavy with wide spacing between their rows of rivets.

The number of girders that it is possible to arrange frequently fixes the spacing, as in Plate I., where only two girders are possible.

Surface to be Stayed. — Having decided on the style of staying to adopt, and on the allowable space to use between rivets, we can locate the outer rivet lines on the tube sheet as follows: —

The tube sheet may be taken as rigidly supported, when bent to an outside radius of 1″, on a line radically distant from its edge $1\frac{1}{4}''$ and, so far as the tube sheet and stay rivets are concerned, no rivets need be placed nearer to this line than they are to each other. Rivets may sometimes be located outside of this limit in girder staying, on account of the position of the through stay rods. In crowfoot staying, Fig. 50, a line drawn radially distant from the shell $a_3 + 1\frac{1}{4}''$ locates the outer rivets of the radial crowfeet.

The lower line of rivets should not be placed nearer than 3″ to the tops of the tubes, for reasons previously given.

Here, as with the tube sheet, the drawing, carefully made, must determine the final arrangement of the stays.

In Plate I., 6″ channel irons were chosen, which fixed the spacing between rows of rivets on them at $3\frac{1}{2}''$ centres; this brings the rivet heads as near the webs of the channel as they can be driven. The greatest distance possible between $\frac{13}{16}''$ rivets has previously been fixed at 4.55″, page 174.

The first line of rivets was placed 3″ above the tubes (the scale on the plate is not correct). Other parallel lines were drawn above these, distant $3\frac{1}{2}''$, 4″, and $3\frac{1}{2}''$ successively, which brought the upper stay rods, when arranged 16″ on centres, low enough to allow access to the boiler. Three rods were now located centrally between the lower two rivet lines with the spacing, allowing the channel iron to rest $1\frac{1}{4}''$ in from the edge of the tube sheet, made equal. This gave 12″ between centres of rods. The upper row of rivets came a little too far down, rendering an angle iron stiffener necessary, as shown.

Size of Stay Rods. — It will be seen that the rods as arranged support part of the tube sheet direct, and part of it through the girders and stay rivets. In general we will assume that a stay rod or a stay rivet supports one-half the surface between itself and the adjacent support. In the case of the tubes, we will assume that the point of support for the plate is located on a line half way between the centres of the tubes and their tops. The support at edge of plate is $1\frac{1}{4}''$ in radially.

The width of the surface stayed by the lower girder is $7\frac{5}{8}''$, and the load per unit of length is

$$7\tfrac{5}{8} \times 150 = 1144- \text{ pounds.}$$

By the theory of continuous girders, for a girder of four equal spans of length l, uniformly loaded with w pounds per unit of length, we have the supporting forces, beginning at the left-hand end, as follows:

$\frac{11}{28}wl$, $\frac{32}{28}wl$, $\frac{26}{28}wl$, $\frac{32}{28}wl$, and $\frac{11}{28}wl$. The bending moments at the supports are 0, $-\frac{3}{28}wl^2$, $-\frac{1}{14}wl^2$, $-\frac{3}{28}wl^2$, and 0.

The maximum load is then on the two outer rods, and is greater than would have been the case were the girder cut at the rods and simply supported.

The load is, $\frac{32 \times 1144 \times 12}{28}$.

The diameter of a steel rod necessary, allowing a working stress of 9000 pounds per square inch, is

$$\sqrt{\frac{32 \times 1144 \times 12}{28 \times 9000 \times 0.7854}} = 1.48''.$$

Before deciding on what diameter to use, the size of the upper rods should be found, and effort should be made to so arrange the rivet lines as to make the rods nearly the same in size.

It can be seen that the upper girder is arranged to receive no support at its ends, which simply carry the plate by means of their action as cantilevers. The outer rivets are located by the principles previously given. The entire area supported by each of the rods is 125 square inches. The diameter of the rod required is

$$\sqrt{\frac{125 \times 150}{9000 \times 0.7854}} = 1.63''-.$$

We will make all the stay rods 1⅝″ diameter and upset their ends, for threading, to 1⅞″. The number of threads to the inch is taken, ten, to insure tight work and save upsetting. It will be found on trial that the stay rod is amply strong even when $\frac{1}{16}''$ has corroded all round.

Steel washers, cupped on their inner surfaces, of a diameter slightly greater than the distance between rivet rows, and thick enough to distribute the pressure of the nut, are placed under the cap nuts on the ends of the stay rods. Cast-iron thimbles, fitting loosely, bring the inside nut beyond the channel. When both nuts are tightened, the plate is very rigidly held. Copper washers are sometimes used with flange nuts. With steel washers, thin copper gaskets are often placed between the outside nut and washer.

The rivets are now arranged to support the plate as evenly as possible. It often becomes necessary to make the spacing less than that calculated, since the surface to be stayed is too great for one rivet and not great enough for two. The spacing around the washers should be symmetrical whenever possible.

The channel irons now only serve to carry the loads, given them by the rivets, to the rods, each rivet staying one-half the surface between it and the next support. The support of the girder is also much stiffer than if it rested on a narrow edge. It is customary to calculate the girder as though it received no help from its connection with the plate. For an exact determination of the stresses in the channel, recourse must be had to the theory of continuous girders. We will, however, be on the safe side for the lower channel if we assume it cut and simply supported at the edges of the stay rods, thus making beams $10\frac{1}{8}''$ long. The maximum bending moment, M, is now determined from the rivet loads, and by the general theory of beams we have

$$M = f\frac{I}{y}.$$

For wrought iron and steel channels we may take f 12,000 pounds per square inch working stress. Substituting, we find a value of $\frac{I}{y}$. A channel iron is now selected which has a suitable value of $\frac{I}{y}$. Or we may, from a table of channels, substitute the value $\frac{I}{y}$, and see if the resulting stress f is permissible.

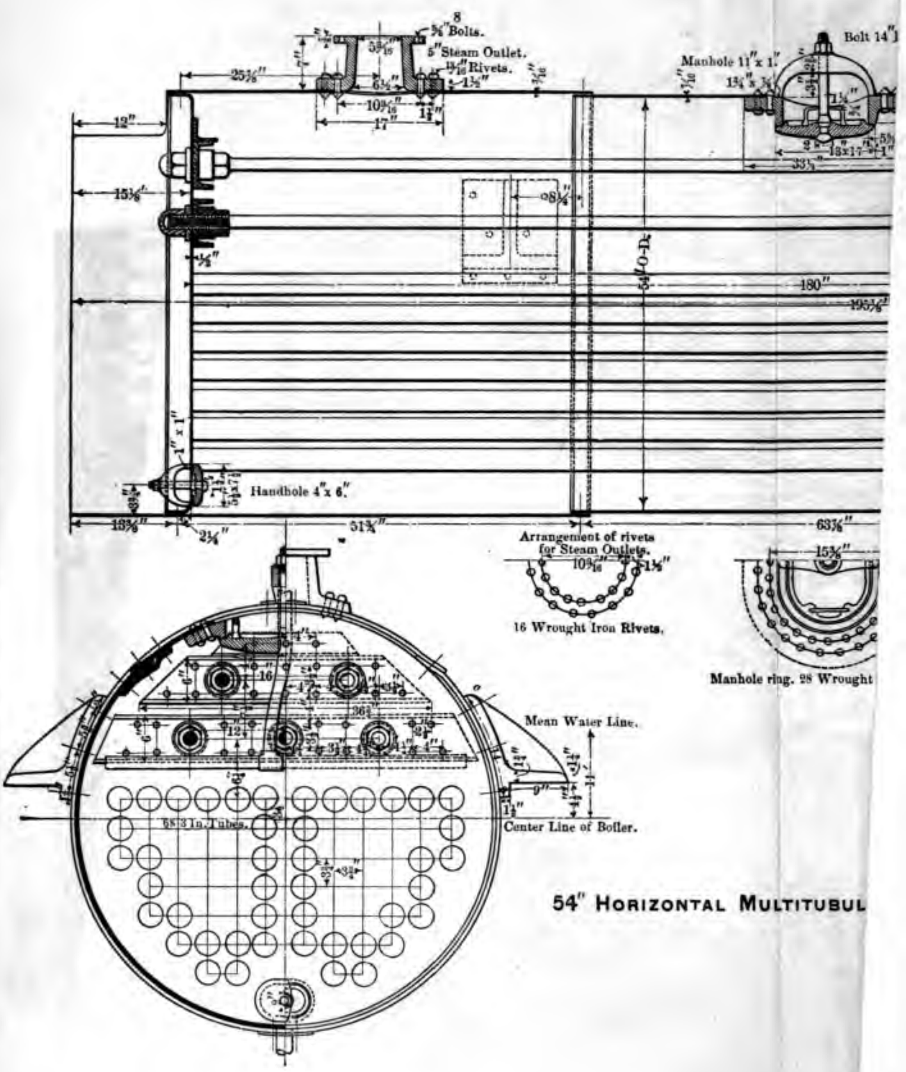
The upper girder may be treated as a beam overhanging its supports. Here the supports may be taken as having a width equal to the diameter of the stay rod. Find M, and proceed as above. The small angle iron may be assumed supported at its ends.

The flat surface of the tube sheet above the hand hole should be braced by two radial stays of $2'' \times \frac{5}{8}''$ iron bent up at the ends to fit the plate. One rivet should be located at each end as near as possible to the bend. A similar stay, arranged for two rivets, is shown at the top of the back head of the locomotive boiler on Plate V.

Supports. — The bracket support, shown in Plate I., is very commonly used. Since the boiler plate is already heavily stressed, it is advisable to spread the rivets, holding the brackets in place, over considerable area. On this account, brackets less than 10″ wide are seldom used for large boilers. The height is also made sufficient to distribute the stress. For boilers up to 16 feet long, four brackets are used; boilers over 16 feet long usually have six brackets. The arrangement is such that they take equal loads and support the shell with as little bending action as possible. For four brackets, the best location is a little less than one-fourth the tube length in from each tube sheet. The height is such that the brickwork may allow one-half of boiler to be exposed to the gases and yet protect the bracket. $4\frac{1}{2}''$ above the centre line is a common location.

To determine the dimensions of the brackets, first calculate the weight of the boiler, assuming it entirely filled with water. Also allow for any added weight, such as covering brickwork, pipes, valves, etc. The resulting weight divided by four gives the load on each bracket. The bearing part of the bracket is usually made 8″ to 9″ long. The resultant load may now be assumed to act 2″ in from the end of the bracket. This load tends to tip the bracket about its top edge *a*. The moment of the lower rivet pulls about the point *a* should equal the moment of the load about the same point. The remaining rivets are put in to take the shear. The rivets used here are usually made larger than those in the joints, $\frac{7}{8}''$ to $\frac{15}{16}''$ being common sizes. Fewer rivets than shown are seldom used, even though they are overstrong. The bracket is calculated as a cantilever, allowing a factor of safety of at least six. The material is usually cast-iron.

The method of procedure in another design would be similar to the above. The details in all cases should be calculated by the principles of applied mechanics, and in all instances where approximate determinations are used care should be taken that the results are on the safe side.



54" HORIZONTAL MULTITUBUL

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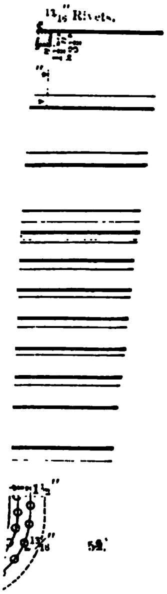

Iron rivets.

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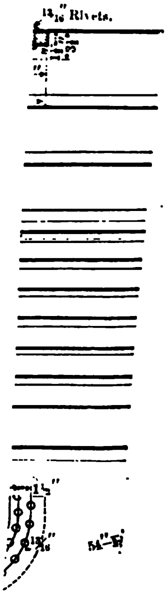

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